

LUNAR DUMP TRAILER

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## ABSTRACT

This report contains the planning, design, and analysis of a vehicle to be used on the lunar surface. This vehicle is designed to carry soil and rocks from a loading point to a dumping point. It will be loaded by another piece of equipment but will dump the soil by itself.

The soil transport trailer is designed to carry approximately ten cubic yards of soil. It was assumed at the start of the design process that the trailer will not be self-propelled; rather, it will be towed by a tractor type vehicle. This towing vehicle will also supply both hydraulic and electronic power to the soil trailer. The trailer will also be remotely operated by a man alongside the vehicle.

The design of the trailer fell into several specific areas. These areas of design included hydraulics, wheels, suspension, frame, and soil bucket. These separate components were then integrated into the final design.

## PROBLEM STATEMENT

### Background

Recently, the National Aeronautics and Space Administration gave Mechanical Engineering seniors at Georgia Institute of Technology the opportunity to design different pieces of equipment whose uses included building and operating a manned lunar station. Our group chose to design a soil transport trailer which would be used to transfer soil and rock from a construction sight to a waste area.

### Performance Objectives

The performance objectives to which the lunar dump trailer must be designed are as follows:

Maximum Speed	30 km/hr
Operational Speed	15 km/hr
Maximum Vertical Slope	30% grade
Maximum Side Slope	15% grade
Hydraulic Pressure	3000psi @ 20 gal/min
Maximum Soil Weight	25,000 moon newtons
Maximum Trailer Height	3.5 m
Maximum Tow Bar Force	75,000 moon newtons
Maximum Tongue Weight	7500 moon newtons
Trailer Capacity	10 cubic yards

### Constraints

Obviously, the biggest constraint on the soil

trailer will be the lunar environment. This environment will challenge the designer greatly; especially in the areas of stability of the vehicle. In addition, the vehicle must be automated as much as possible due to the limited mobility of the personnel around the vehicle.

Safety will also constrain the trailer in some respects. For example, if the vehicle needed to be serviced or repaired, the man doing the work should not have to worry about sharp points or catches on which he could rip his protective suit. Such an accident could be catastrophic. Thus, when designing the vehicle, our group wanted the vehicle to be safe and easy to work with in a restricted environment, even though in normal operation no physical contact with the vehicle by a human would be necessary.

## LUNAR DUMP TRAILER

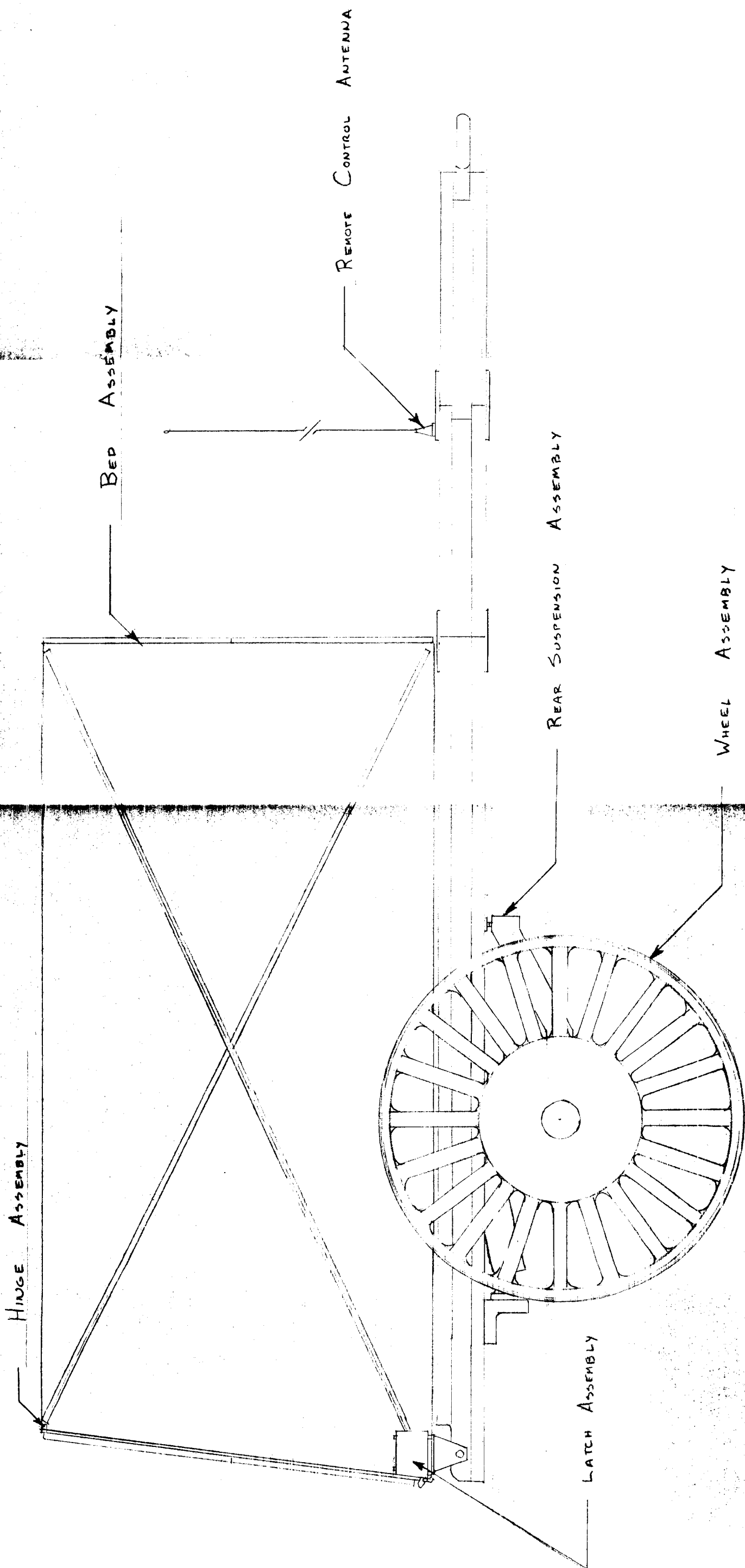
In the following section, each of the major components and their corresponding minor components will be described at length. Assembly and interaction of the respective parts will also be examined.

### Bed

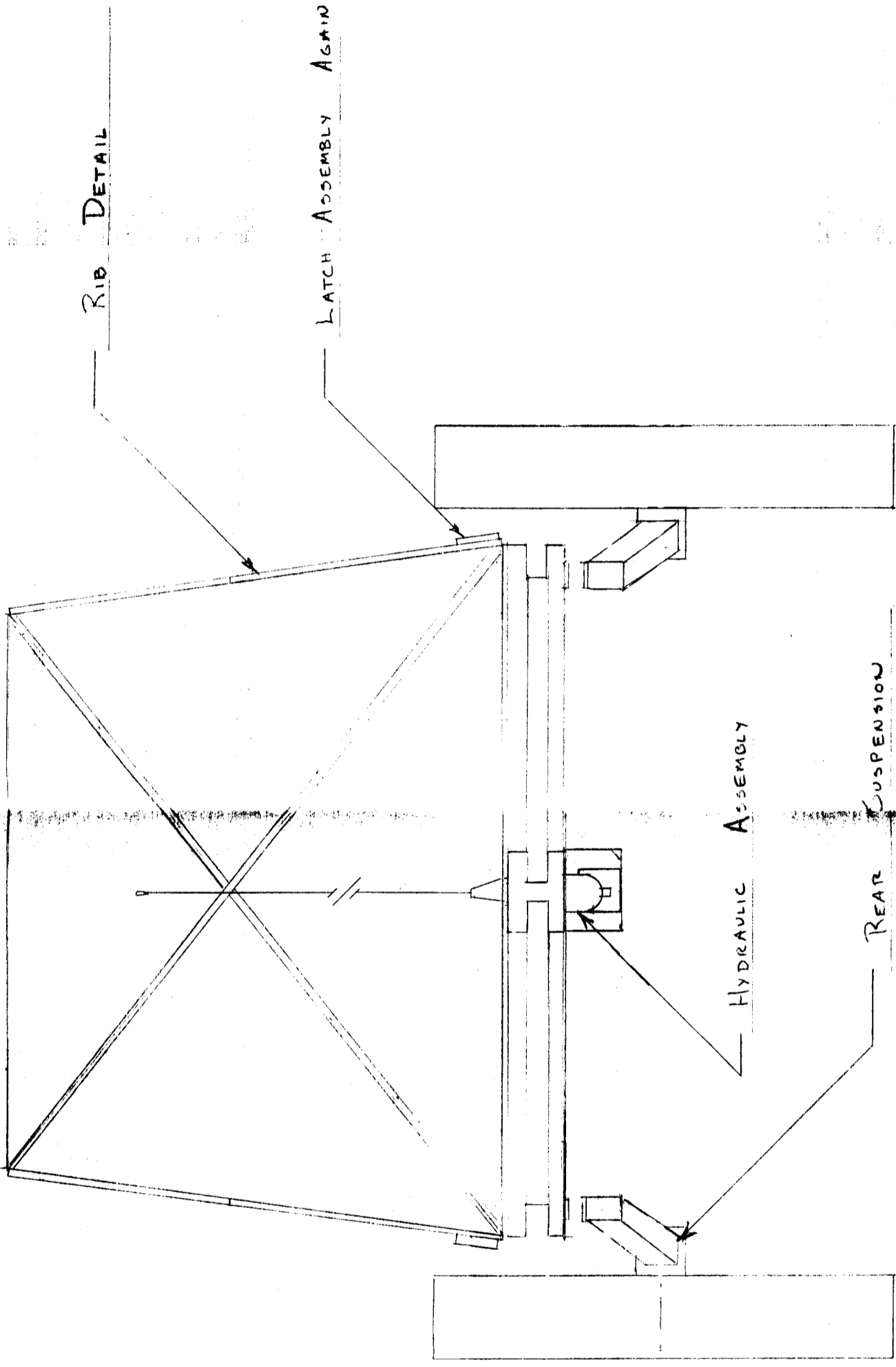
After debating several design ideas for the bed, a basic rectangular bed with three slightly sloped sides was chosen. The slightly sloped sides (see figures 3, 4, & 5) would both lower the center of gravity (increasing overall stability) and help to control dust release from the top. The sloped sides (7.6 degrees) would not at all hinder ease of loading or substantially cut into the overall volume.

The bed is to be made out of an alloy of aluminum which gives a superb combination of strength and toughness. In addition, without any loss of ductility, both strength and toughness increase with decreasing temperature. The alloy chosen was Al2024-T86. This alloy is as strong as steel and masses only one third as much. It also displays less shrinkage in the wide temperature range found on the moon.

For added support and to reduce deflection of the walls, T-beams (see figure 4) are to be welded along the diagonals of all the faces, increasing the moment of inertia for the walls by a factor of four. Also, the



TRAILER SIDE VIEW	
SCALE: 1:40 CM	APPROVED BY:
DATE: 1 JUNE 85	DRAWN BY A. BADE
	REVISED
DRAWING NUMBER 1	



TRAILER		FRONT VIEW	
SCALE: 1:40 cm	APPROVED BY:	DRAWN BY A. BADE	
DATE: 1 June 85		REVISED	
		DRAWING NUMBER	
		2	

bed is rounded at the base, this gives added support and permits greater ease in unloading.

The last major components are the brackets which hold and allow the rear door of the bed to swivel and release the soil (see figure 6). They are bolted for ease of removal in case of failure or wear. The brackets are bolted in only one place, the other anchor is provided by a stub sticking up from the bed. This allows easier access to the one bolt, with no loss in beneficial properties. These brackets can be easily cut and bored to their proper dimensions, and will be made from the same material as the bed.

The dimensions of the rear door are the same as those of the outside of the beds edges. A round rod is welded to the top of the door, a rod which will swivel easily in the brackets on either side without any need for lubrication or close tolerances.

All of these components are simple and proven reliable. They are well designed for safety, reliability, and weight requirements, although at the expense of economy.

#### Frame

The frame is designed using an I-beam construction. From the main frame, the beams expand at a 30.25 degree angle to the hitch. The main members, consisting of the beam under the bucket and the 30.25 degree extensions, run under the outside edge of the bucket (see figure 7).

Just past the point where the hydraulic support cross members attach, the I-beams become inverted T-beams so as to allow attachment of the bucket to the frame. The trailing edge is rounded to allow the bucket to rotate. All connections of cross members are composed of plates and bolts on both the top and bottom flanges of the I-beams. All components--beams, plates, and bolts--are made of Al2024-T86, an aluminum alloy.

The need for a hitch bracket to connect the hitch and the frame is obvious. The hitch is made of the same alloy as the frame and bucket for strength, low mass, and conformity with the rest of the vehicle. The bracket is connected with the same type bolts as used on the frame. The design of the bracket resulted from the need to join the two pieces of the frame which meet at a 60 degree from the horizontal. Thus, the bracket needed 60 degree flanges on the bottom portion to accommodate the two beams from the frame. With the hitch being parallel to the vertical, the bracket takes the given shape (see figure 7).

#### Suspension

A pure trailing arm suspension was used, consisting of two independent sides, each a mirror image of the other. Each side is made of a trailing arm, an axle, a coil spring, a shock absorber, and various connecting pins and bolts (see figure 8). The trailing arm connects the axle to a point on the frame forward of the

wheel. The spring and the shock absorber are connected in parallel (coil spring over shock) from a point near the trailing arm's mounting point to a point inboard of the wheel (see figure 9).

Since all of the lunar dump trailer's mass is supported by the spring, the trailing arm experiences only slight torsion. This torsion then transmits a shear force to the pin that connects the arm to the frame.

Most of the mass saved in this suspension is due to its short axles. The axles were analyzed for both shear and bending. They did not have to be analyzed for wear because they do not rotate.

#### Wheel

The wheel is a one piece forging made from Al2024-T86 alloy. The outside diameter is 1.4 meters, and the hub diameter is 0.64 meters. The wheel has 20 spokes spaced 18 degrees apart, and the spokes are 6.0 cm in diameter. The wheel is attached to a threaded axle. The nut is held in place with a cotter pin. This arrangement allows for quick removal of the wheel from the suspension, and easy installation of a replacement wheel.

#### Hydraulic Components:

Cylinder. The cylinder used is a double acting cylinder with cushioning on the push and pull stroke. A double acting cylinder has fluid pressure applied to the

movable element in either direction. Cushions slow a cylinder's piston movement just before it reaches the end of the stroke. A cushion consists of a needle valve flow control and a plug attached to the piston. As a cylinder piston approaches the end of its travel, the plug blocks the normal exit for a liquid and forces it to pass through the needle valve. At this point, some flow goes over the relief valve at the relief valve setting. The remaining liquid ahead of the cylinder piston is bled off through the needle valve and slows the piston. The opening of the needle valve determines the rate of deceleration.

Flow Control Valve. The function of a flow control valve is to reduce a pump's flow rate in its leg of a circuit. These valves provide precise control of flow and shut off in one direction, and automatically permit full flow in the opposite direction. A two step needle allows fine adjustment at low flow by using the first three turns of the adjusting knob; the next three turns open the valve to full flow, and also provide standard throttling adjustments.

#### Solenoid Operated Directional Control Valves.

A directional control valve consists of a body with internal passages which are connected and disconnected by a movable part. The directional control valve used in the lunar dump truck system is a 4 way - 3 position - centered solenoid operated directional control valve

with a float center. When an electric current passes through a coil of wire, a magnetic field is generated. This magnetic field attracts the plunger and pulls it into the coil. As the plunger moves in, it contacts a push pin and moves the directional valve spool to an extreme position.

Pilot Operated Check Valve. A pilot operated check valve allows free flow in one direction. In the opposite direction flow may pass when pilot pressure unseats the valve's movable member. The pilot operated check valves will block the flow out of the cylinder. The load will be locked as long as the cylinder seals remain effective.

Quick Disconnects. The quick disconnects allow the trailer to have a means of disconnecting the hydraulic lines from the towing vehicle. These devices will be located near the end of the towing bar and can be operated manually.

#### Door Latch

Two latches are used on the lunar dump trailer to keep the bed door shut. They are attached to the bed at the lower rear corners (see figure 17). The door is allowed to open when a solenoid is energized and rotates the latches to the open position. After the door has been opened, the solenoid is released and the latches move back to their original positions. Once the bed is lowered, the door will swing back into place and the

latches will lock the door shut.

#### Remote Control

The lunar dump truck is controlled by a remote control device that uses a radio signal of 27.145 Mhz as a carrier. An amplitude modulated signal is transmitted by the person in charge to the lunar dump trailer and received by a receiver via an antenna. The remote control unit will control two functions of the lunar dump trailer. The raising and lowering of the bed and the opening and closing of the bed door.

The bed movement is controlled by a hydraulic valve which operates the hydraulic ram. The remote control will control this device by the use of a silicon controlled rectifier (SCR) or thyristor. When the proper signal is received, a small charge is sent to the thyristor by the remote control unit, which then charges a relay. The relay then connects lines beta and gamma (see figure 18) which allow power to flow to the hydraulic control valve, causing the bed to rise. The connection of lines alpha and gamma (see figure 18) on the relay allow power to flow through a resistor. This resistor imitates the needs of the hydraulic control valve. This is done in order to have at all times the proper amount of amperage division by the parallel circuit. Once the signal is discontinued, the thyristor will discharge the relay and cause the valve to reverse, resulting in the lowering of the bed. The latches are

controlled by solenoids. The remote control unit controls the solenoids in the same manner as the hydraulic valve.

The remote control unit is contained within a box attached to the frame. It is located under the bed in a corner to allow the maximum amount of protection. The antenna is connected to the towing bar. The power supplied by the trailing vehicle is attached to the lunar dump trailer by quick disconnects.

## DESIGN AND DEVELOPMENT

In this section, the reasons for design decisions are described. It contains all assumptions made in the design process, and it will state such things as safety factors, modes of failure (by shear, bending, torsion, etc.), and how each of these affected the design the component designs.

### Bed

#### BED DESIGN

In designing the actual shell of the bed, the soil was presumed to act approximately as a fluid at the geometric center of the walls. By finding the density of the soil and integrating along the depth of the trough, the maximum moment possible at the bottom of the wall, excluding the helpful support from the other walls, was calculated and used to compute the minimum thickness needed along the bottom to support this moment. A safety factor of two and large fillets were added to increase the factor of safety to a value of at least five. The safety factor is even higher on the smaller front and rear walls.

Since unpredictable stress concentrations would be extremely detrimental to a highly reliable dump trailer bed, methods of forming the bed such as stamping,

rolling, etc., are quite unacceptable. Stress concentrations through processing plus additional stress concentrations due to high variations of temperature make casting one of the best solutions. It would completely eliminate the need for bolts, bolts which might not even fit due to shrinkage at low moon temperatures. Casting the entire bed would be extremely expensive and difficult due to the thin walls. Die casting is the best way, but a hot cast and extensive well systems are necessary to avoid early freezing of the molten aluminum. This cast forms the basic shell of the bed.

The T-beams were added to decrease wall deflection and to increase the safety factor by two. They interfere with nothing and increase the total weight of the trailer by negligible amount.

The bottom carries a safety factor of three (assuming no supporting members), and it can withstand cracks caused by large rocks falling from a distance of 2 meters.

There are two sets of hinges for the bed component: (1) the hinge for the swivel at the top, and (2) the heavy duty swivel upon which the entire bed swivels in unloading. All rods and inner holes in the brackets for the tubes are coated with teflon to reduce friction and increase durability, and are mounted with bolts for ease of removal. The design of the brackets and

corresponding rods was based on the analytical result that the shear force would be the dominant source of failure. The swivel at the top was made to account for failure due to the two components of force: (1) the pressure force out, and (2) the weight of the door hanging on the swivel. A safety factor of at least eight was added. The swivel for the entire bed was made to withstand the entire load if it had to, times a safety factor of five. The swivel bracket was chosen from a commercial catalog, and the match pin has an even higher safety factor.

#### Frame

The frame of the lunar dump trailer is the unifying item for the suspension, the bed, the hydraulic ram, and the towing vehicle. The frame was designed to support each component based on a worst case condition.

The material chosen, Al2024-T86, was picked because of its high strength to mass ratio. The frame is constructed with an I-beam composed of members 4.75 cm by 10.0 cm, formed by a continual cast. However, since the bucket is designed to mate with a flange, the top plate of the I-beam is cut off, with the use of a laser, near the end of the frame. The laser cutting will alleviate any stress concentrations which would result from other types of cutting. Therefore, the tail end of the frame members are effectively inverted T-beams. Additionally, all calculations were made based on the

T-beam since this is the weakest portion. A safety factor of eight is obtained when the entire loaded bed acts on the single beam. Therefore, under normal conditions, the safety factor for the T-beam bed supports is around sixteen. Obviously, the safety factor for the rest of the frame is much greater than eight.

The rear cross member is designed to support the forces exerted by the hydraulic ram. An L-beam is attached to the lower side of the beam to allow for the attachment of the ram.

The bolts and plates used to connect the beams were again designed on a worst case basis. The shear developed by the ram force was found to be the limiting factor for bolt design. The analysis was carried out on only one plate attached to a member. Therefore, since a plate is used on both the upper and lower flanges of the I-beams, the factor of safety is increased by a factor of two. An Hewlett Packard HP-41 program, BS (bolt shear), was used to calculate the force and angle on each bolt. A pattern of four bolts on each beam was used. The bolt size was determined to be 1.25 in. in diameter and 5.0 in. in length. The plates are 10 cm by 20 cm and 4 cm thick.

The towing beam is attached to the frame at the end of the side support I-beams, angled at 45 degree angles. A round, open hole hitch is used. All connections for

electrical and hydraulic connections are located along this tow bar.

### Suspension

The type of suspension used on the lunar dump trailer is called pure trailing arms. Pure trailing arms have the advantages of low upsurge mass, low total mass, zero camber change, right to left independence and overall simplicity. A Hotchkiss suspension (leaf springs and shocks) was also considered but was eliminated because of high mass.

The springs were designed by first selecting a spring constant that took into account expected forces and desired suspension travel. Then, the spring was sized using a program on the Hewlett Packard HP-41CX.

After the spring constant was specified, the damping coefficient was selected by using another program on the HP. A commercial damper was selected from the ACE catalog in the library.

The wheels were placed just behind the center of gravity of the lunar dump trailer to reduce the upward and downward forces on the tongue. They were also spaced well outboard of the bed to increase stability.

### Wheel

The basic requirements for the design of the wheel are to: be able to support a 13,400 moon newton load over small obstructions with minimal rolling resistance, to contribute to a reasonable ground

clearance, and to be extremely reliable. The wheel had to be extremely reliable since the replacement and repair of the wheels would be difficult in the lunar environment. Another associated requirement, the primary requirement of the suspension system, is that the wheel transmit the lowest possible forces and accelerations to the bed in order to minimize the amount of motion experience by the cargo.

The last requirement controlled early design decisions about the wheel. The goal in the beginning was to design a wheel that was an integral part of the suspension. There are spring and dampening properties built into the wheel itself in addition to those in the frame suspension. There were several problems with the earlier designs that employed internal wheel suspension. The first design (see figure 10) was composed of hoops of spring steel enclosing a doughnut of a dampening material. This material would help absorb impacts before they affected the trailer. The biggest problem with this design was finding materials that could serve as a suitable dampers and also survive the extreme temperature ranges expected on the moon. Another problem was the severe abrasion of the riding surfaces and dampening material. With such a complex construction involved, the wheel would have very low reliability, and therefore a short life. Along with poor reliability, this design also had to offset high

rolling resistance with poor fatigue resistance.

The second design consideration was an attempt to construct the wheel on a "live floating hub" concept. As seen in figure 11, the outer rim of the wheel is rigid, and the struts are composed of a series of springs and dampers. The hub is allowed to move and thus absorb some of the impact forces before they affect the trailer. Several problems, similar to those that plagued the first design consideration, exist in this design. The first concerned stability. Even with the crossed strut construction, there was a concern that the rim would not be stable around the hub, especially over rough terrain. The higher rolling resistance experienced by the eccentric center was the second problem. Other problems concerned reliability, complexity of construction, and problems with replacement of damaged or defective struts in a lunar environment.

With so many problems associated with a live wheel, passive wheel designs were considered. One constraint was to not use material that decomposed in the high radiation environment of the moon. This led to the decision to use an all-metal wheel. Aluminum was chosen for its better strength to mass ratio. A spoked design was chosen over a solid wheel for the reduction in mass. A rigid wheel also would show superior reliability and longer life, characteristics that are highly desirable

in a hostile environment.

#### Hydraulics

The initial design process began with the reading of several books about hydraulics, particularly the Fluid Power Handbook (put out by Hydraulics and Pneumatics), and two books supplied by Parker Fluidpower called Design Engineers' Handbook and Industrial Hydraulic Technology. Several pamphlets about construction equipment were also studied. The initial design considered was one consisting of two hydraulic rams. Calculations were done to determine the force required to lift the bed when applied at different positions and bed angles. The bore and stroke required were then calculated. After looking at the size of piston required, it was determined that one cylinder would perform the job adequately and thus eliminate the number of components and phasing problems that are involved when two cylinders are used. Once all of the parameters for the hydraulic system were known, a hydraulic sales and consulting firm was contacted in order to determine cost and availability of the hydraulic components. Using this information as well as the information gathered from the books, the hydraulic system was designed.

#### Door Latch

The designing of the latch took many considerations. The first, and certainly not the least,

is the fact that it is remotely controlled. This requires either a wire pull mechanism or something electronic in nature. The wire mechanism was ruled out due to wear and possible external damages. Therefore, an electronic method was employed. A solenoid pushes a latch similar to that of earth dump trucks. The latch is then returned to its original position by the solenoid so that the door would lock in place when it rotated back to its original position.

The failure analysis was calculated on the assumption that lunar dust and rocks act as a fluid. This incorporated a built in safety factor since the lunar soil does not act as a fluid. On top of this built in factor, a safety factor of six or better was used through out the calculations. This leaves room for abuse of the trailer. Also, teflon was used to reduce friction on frictional surfaces. This was the material best suited to conditions of the lunar environment.

#### Remote Control

The design of the remote control apparatus had the initial considerations that it be wireless and easily operated at a safe distance from the lunar dump trailer. Therefore, a remote control unit that employed radio waves for control was designed. An antenna is used for receiving the radio waves and sending a signal to the remote control unit.

The remote control device consists of a box

containing the electronic circuitry with quick disconnects on the outside for easy replacement. The electronics contained in the box must first reduce the 100 amperes coming from the towing vehicle to levels needed by the lunar dump truck equipment. This is done with a parallel circuit and required resistors to be used at all times to insure that the device maintains a constant current drain. To switch between modes, relays are placed in the circuit. The remote control device controls the relays via thyristers which switch large amounts of power with little power input. This is the circuit used in the design of the lunar dump trailer and can be seen in figure 18.

## COST ANALYSIS

Production unit cost for each dump trailer will remain high for the initial run. For several average sized moon bases, it will require about 30 vehicles to move sufficient earth around for construction. The cost of casting 30 beds would be extremely expensive, but forging and welding it together would also be plausible, although this too would be expensive. The forging or casting equipment needed for the bed would have to be made and operated at high cost. In addition, forging equipment to construct the wheels and spokes would have to be made. Only about 60 wheels, plus about 10 extra, would be made with a cost of about \$17,000 per wheel (including bearings). Cost, especially transportation costs, would eventually go down when a production site begins on the moon itself.

Making quality parts for so few units with high reliability is very expensive. Equipment necessary to make the bed and wheels will have a high initial cost and units cost, but other parts can be easily bought from commercial manufacturers at a comparatively low price. Only testing and extensive research could lower the high price tag.

The parts list for the trailer design is as follows:

BED:

Main Frame

Materials	\$6000
Casting & Machining	\$37000
Total	\$43000

Door

Materials	\$800
Casting & Machining	\$8000
Total	\$8800

Miscellaneous

Bolts, Pins & Brackets	\$800
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Total for Bed	\$52,600
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FRAME:

I-Beams

Materials	\$10000
Casting & Machining	\$15000
Total	\$25000

Miscellaneous

Bolts & Plates	\$500
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Total for Frame	\$25500
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SUSPENSION:

Trailing Arms

Materials	\$500
Bending & Machining	\$3000
Total	\$3500

Axles

Materials	\$100
Machining	\$1000

Total	\$1100
Springs	
Materials	\$300
Forming	\$2000
Total	\$2300
Shock Absorbers	
Purchase Price	\$1500
Miscellaneous	
Bolts and Pins	\$1000
<i>Total for Suspension</i>	<i>\$9400</i>

#### WHEEL:

Spoked Wheel	
Materials	\$1500
Forging	\$10000
Total	\$11500
Miscellaneous	
Bearing	\$500
Hold Nut	\$500
Total	\$1000
<i>Total Wheel Cost (x2)</i>	<i>\$24,500</i>

#### HYDRAULICS:

Cylinder	\$930
Valves	
Flow Control	\$54
Solenoid Directional	\$239
Pilot Operated	\$223
Total	\$516

Mounting Accessories

Mounting Plate	\$12
Sub Plate	\$38
Eye Bracket	\$12
Pivot Pin	\$4
Total	\$66

*Total of Hydraulics*                      *\$1512*

LATCH:

Forged Latch

Materials	\$100
Forging & Machining	\$1000
Total	\$1100

Miscellaneous

Cover	\$100
Solenoid	\$50
Bolts & Nuts	\$25
Total	\$175

*Total of Latch*                      *\$1275*

REMOTE CONTROL:

Electronics

Remote Control	\$40
Resistors	\$100
Relays	\$50
Thyristors	\$20
Board	\$75
Wire	\$300
Disconnects	\$100
Total	\$685

Miscellaneous

Box	\$75
Antenna	\$150

Bolts \$20

Total \$245

*Total for Remote* \$930

***Total for Trailer*** **\$115,717**

## HAZARD ANALYSIS

The possible shortcomings of the design and the consequences of any error in the designs will be examined for each component. We will also look at how poor design will effect the safety of the operating crew. Concerns of the designers for certain components, along with any remedies, will also be looked at within this section.

### Bed

The actual bed itself poses no real hazard to its operators. Buckling in the walls would be apparent long before any failure occurred. If any chance for failure exists, it is in the rear door upper latch. The round bar upon which the door swivels must be carefully welded on under vacuum conditions to prevent failure. Going straight up very steep inclines might also cause failure, but the tractor could probably not even get enough traction to go up inclines this sharp (greater than 40 degrees).

The only real problem with the bed is that any metal, at such low temperatures, is extremely brittle. Great care was taken to select an aluminum alloy with very little change in ductility, even with a temperature range between 340 K and 30 K. But, large rocks dropped from a height of 2 meters might cause cracks in the walls and in the bottom which might eventually lead to

failure. Operators should avoid picking up and dropping large rocks into an empty bed. The safety factor in the bottom should easily be able to handle such a situation anyway.

#### Frame

Since the frame is the unifying member of the entire design, any failure in the frame will result in the disablement of the lunar dump trailer for any useful purpose. However, if a bolt on a connecting plate were to fail, it could be replaced thus restoring the lunar dump trailer to service. This is possible since the bolts were designed so that failure of one will not result in the failure of the others. Routine inspections should spot any troubles and cure them before any catastrophic failure occurs.

#### Suspension

Since the lunar dump trailer and tractor are remote controlled, no suspension failure is expected to cause injury or endanger human life in any way. Failure of the suspension may however, result in down time for the lunar dump trailer.

The area most likely to fail on the suspension is the bearings. They are sealed against the moon's environment but they have not yet been tested in such an environment. They are also the only part of the suspension that is subject to wear.

The damper is also subject to failure because the

damping constant gradually decreases over time. The characteristics of the suspension will slowly move from overdampened to underdampened. This is not expected to cause a problem before the suspension is changed.

Due to the temperature extremes on the moon, structural pieces may fracture when given an unexpected, sharp jolt. This could happen if the lunar dump trailer was moving at high speed and the operator was unable to see rough terrain because of a dust cloud.

In each of the above mentioned cases, the very worst thing that could happened to the suspension is that it would have to be replaced. This can be done relatively easily since each side is connected to the frame at only two places.

#### Wheel

These are several areas of failure that have been recognized and designed for in this wheel. These are:

1. Failure by buckling of the spokes
2. Failure by buckling of the outside rim
3. Failure by fatigue of the spoke hub  
or spoke rim interface
4. Bearing failure.

The first failure mode can be caused by several things. Overloading the dump trailer in significant excess of its design load will greatly increase the chance of the spoke buckling while the trailer is moving. This is due to the load per spoke increases

with increased axle load. Greatly overloading the trailer could result in the spokes buckling while the trailer is stationary.

The spokes could also buckle under excessive impact loads, which would be caused by higher speeds over rough terrain. At normal operational speed, the trailer wheels would be able to go over an obstruction as large as half a meter tall without failure. High speeds will increase the forces the spoke would experience to dangerous levels, and increase chance of buckling.

The consequences of buckling would be dependent on the severity of the failure. If the failure of the spoke was severe enough to also result in the buckling failure of the outer rim, the wheel could be considered destroyed. If, on the other hand, the buckling of the spoke only resulted in the deflection of the spoke and the rim was left intact, the wheel could still be used if certain precautions were taken. The first precaution would be to straighten the deflected section by some means, such as hammering or pressing. The wheel could then be used again in a lighter duty situation, such as smoother terrain and a reduced cargo capacity. These precautions are necessary because the deformed spoke will be weaker than the other spokes, and continued normal use will result in the eventual destruction of the entire wheel. This procedure is included so that lightly damaged wheels can still be used on the lunar

base in some manner. This is important because replacement parts for repairs will be scarce and expensive on the moon.

The second mode of failure, the buckling of the outside rim, will most probably be followed by failure of one or more spokes in the immediate area of the rim failure. Failure of the outside rim in the above manner should be considered catastrophic failure, and will always result in the destruction of the wheel. It should be noted that very high impact forces will be required for a failure of this type, and that normal operation will not result in this type of failure unless very unusual circumstances are present.

The third mode of failure, fatigue failure, will probably be the rarest type of failure associated with the wheel itself. Due to design results from other parts, the fatigue safety factor is 230 after a use of 110 Mkm. Fatigue failure should not be completely discounted, however, and regular inspections of the wheels for fatigue cracking of the spokes should protect against fatigue failure. These inspections should be scheduled once every 250,000 km of use, or every 2 years, whichever comes first.

The fourth form of failure, bearing failure, will be the most common anticipated failure associated with the wheels. The use of sealed bearings in the wheels should eliminate problems caused by lack of lubrication

and by dust and grit entering the bearing surfaces. The admittance of sand and grit into the bearing will quickly result in bearing failure, and this is the reason sealed bearings are used. Fatigue failure of the bearings is the other method in which the bearings will fail. It is recommended that the bearings be replaced yearly to protect against their failure. If the circumstances warrant it, the bearing could be returned to earth or to a space station for inspection and rebuilding. Inspection would not be advisable on the moon because of high dust and dirt content, which would contaminate the bearing surfaces. Bearing failure will result in the inability of the trailer to be rolled, since only one wheel will turn. This could result in a dangerous situation if the trailer stops in a heavily traveled region, so bearing failure should be prevented by maintenance as much as possible.

#### Hydraulics

The hydraulic cylinder used is a heavy duty cylinder and should provide millions of strokes without failure. Because the system incorporates pilot operated check valves, any loss in pump pressure will result in the cylinder being locked into position, thus eliminating the bed from falling if power is lost in the upright position. The hydraulic ram is cushioned at both ends. This slows the piston movement at the end of the stroke and prevents damage to the cylinder by having

it slam home.

#### Door Latch

The foreseeable hazards of the door latches are limited. The latch was designed with a safety factor of at least six. This high safety factor will prevent any real danger to the user. The possible problems that can occur with the latch assembly are in the solenoid. The solenoid can either lock open or closed.

If the solenoid does not operate, the door will not open for the dumping procedure. This will result in the entire load being exerted upon one pin. This will cause one of the pins to shear. Therefore if the operator sees that the door is not opening he should immediately halt the dumping process.

If the solenoid does not return to its original position, the door will lock shut by the other latch. This will result in all of the load being exerted upon one latch. With the safety factor that is included, this will not cause failure of the latch. However, the door may bend due to the absence of support in one corner. Therefore the operator should visually inspect the latch assembly to insure that they have both returned to their original positions.

#### Remote Control

##### REMOTE CONTROL HAZARD

The hazards that are associated with the remote

control unit will occur when an electrical component malfunctions. The malfunction will result in either the hydraulic lift or the door latch solenoids not working. The results of either of these failures are discussed previously in their respective hazard analysis.

A remote control unit that has failed will have to be fixed in a space station or other confined area. Therefore, the remote control unit is designed for easy removal. Four screws would be removed and the electronics would be disconnected via quick disconnects.

## OPERATING INSTRUCTIONS

For safe operation of the lunar dump trailer, the following parameters should not be exceeded:

1. Maximum speed of 30 km/hr over smooth terrain.
2. Maximum speed of 15 km/hr over rough terrain.
3. Obstacles greater than 0.8 meters tall  
should not be driven over.
4. Obstacles greater than 0.6 meters tall  
should not be transversed by the wheels.

The dump trailer will transverse a slope of 20% grade safely. The trailer should not dump it's load on a transverse slope in excess of 10% grade due to the dangers of tipping over. The trailer may be driven up or down grades of 30%.

It is important not to overload the lunar dump trailer over the top of the bucket. Doing so could result in a load in excess of the design load. This could lead to damage being inflicted upon the vehicle.

To perform the dumping procedure, the operator will use the remote control device. He will first trigger the latches to open and then turn on the hydraulics to raise the bed. The vehicle should be moved slowly during this process in order to prevent the dirt from piling up in the trailer. Once the load is deposited, the user will release the latches and then lower the bed by turning of the hydraulics. If the door does not open during the dump sequence, he should immediately stop

dumping and inspect the latches.

## MAINTENANCE

Although the lunar dump trailer should operate safely for several years, certain parts will require regular maintenance and inspection. The following schedule provides periodic maintenance periods for certain tasks to be performed. (note: all time periods given are in terms of earth days, not lunar days)

### Weekly

1. Remove latch cover plate, inspect latch assembly and remove accumulated dust
2. Visually inspect hydraulic ram and lines for signs of leakage or damage

### Monthly

1. Inspect hinge brackets and bucket hinge brackets for wear of teflon coating
2. Inspect remote control wiring for damage or wear
3. Inspect suspension attachments points, spring, and damper for wear or damage
4. Inspect quick disconnects for wear or damage

### Semi-Annually

1. Inspect frame and bucket visually for cracking or other signs of wear
2. Inspect hydraulic ram and control valve for signs of wear and damage
3. Inspect remote control unit for signs

of heat damage or wear

4. Pull wheels and replace bearings

Annually

1. Check wheels, rear suspension, and hydraulic ram mount for fatigue cracking
2. Replace remote control unit
3. Inspect internals of ram, hydraulic control valve for wear
4. Replace suspension spring and dampeners
5. Visually inspect entire trailer for cracking or other signs of wear
6. Replace latch assembly

## CONCLUSION

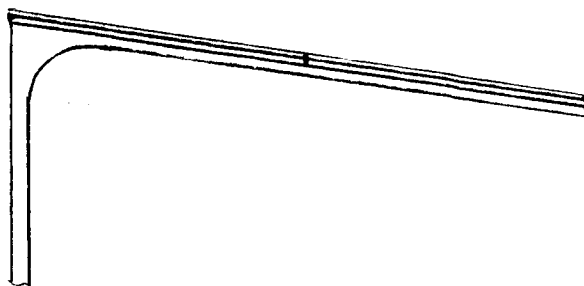
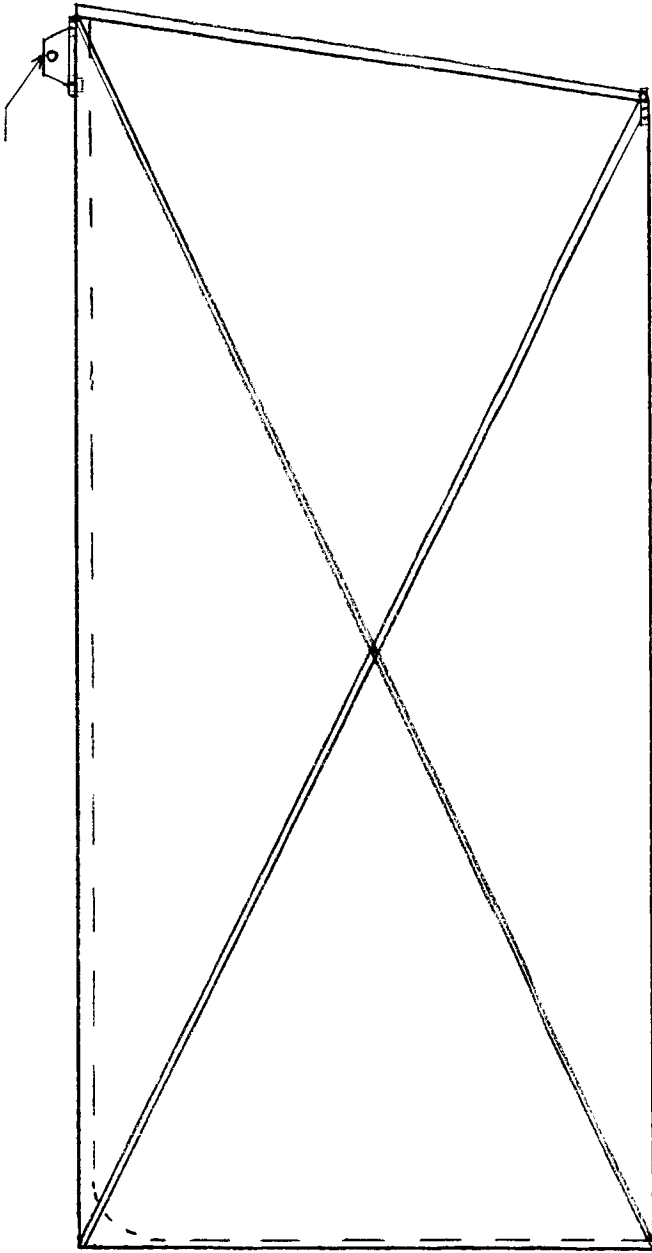
The dump trailer was designed with several basic principles in mind. The first principle was that cost was virtually no object. The cost of transporting the trailer to the moon and the cost of a failure are high in comparison to the cost of building the lunar dump trailer, therefore the building cost can be neglected.

When determining safety factors, both avoidance of failure and weight savings were taken into account. Operation of the lunar dump trailer does not pose a threat to human life since it is remote controlled. A failure would, however, be very expensive in downtime and this expense must be balanced with the added expense of transporting and towing a heavier trailer.

A third principle was that the machine had to be simple to operate by a man in a spacesuit in a dust environment. This also includes maintenance. Small or hard to get at fasteners were avoided.

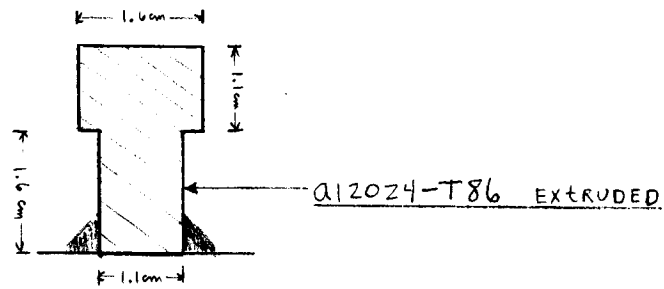
FIGURE 3

BED: SIDE PANEL  
( $Z'' = 1m$ )



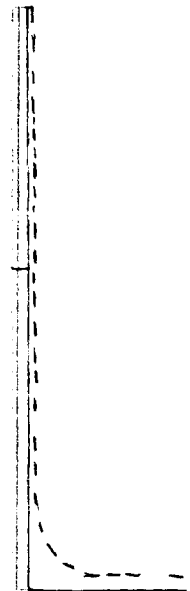
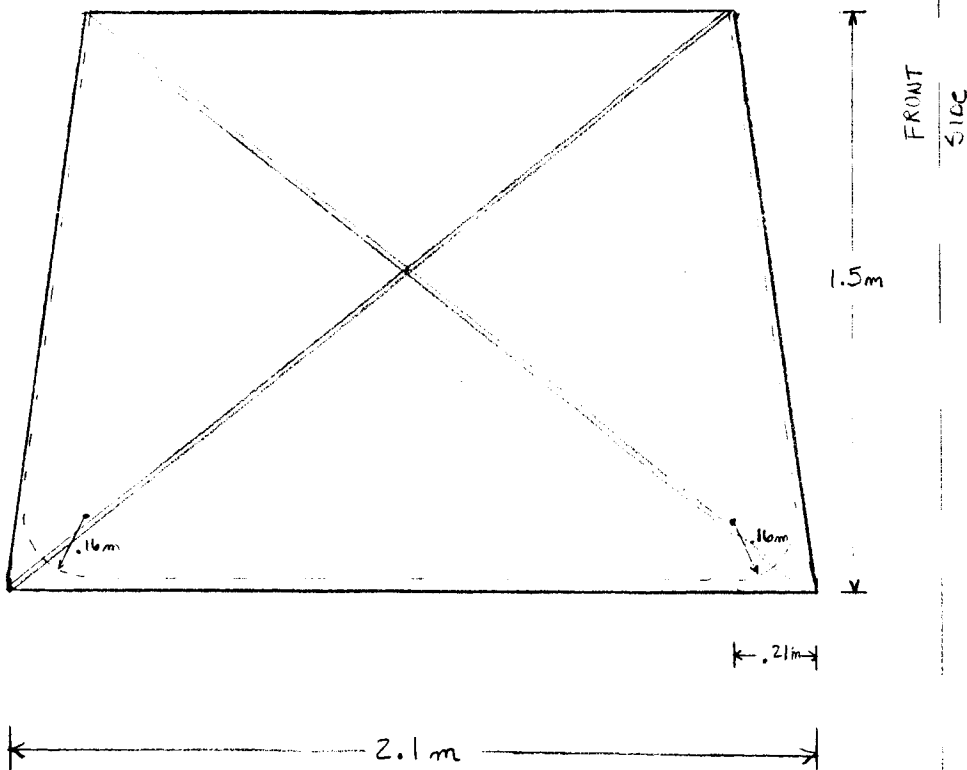
FRONT SIDE:

CRISS-CROSSED T-beam:  
(actual size)



b.

(2" = 1m)



BED: REAR GATE  
(2" = 1m)

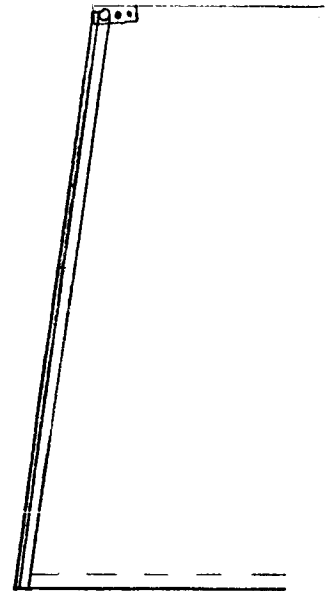
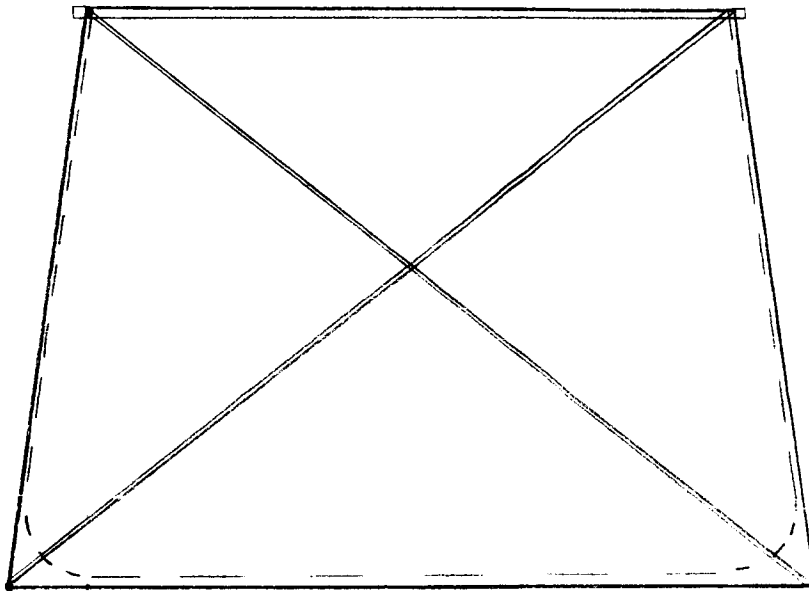
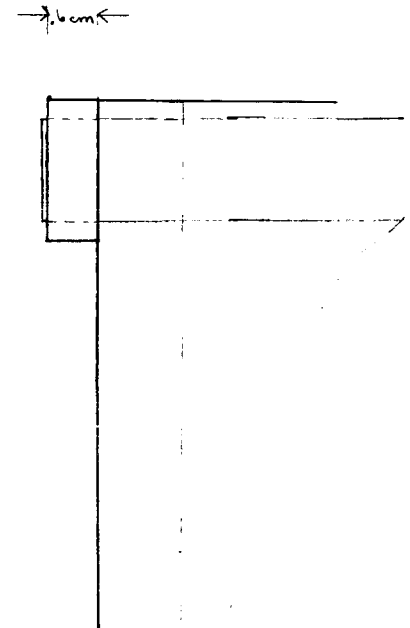
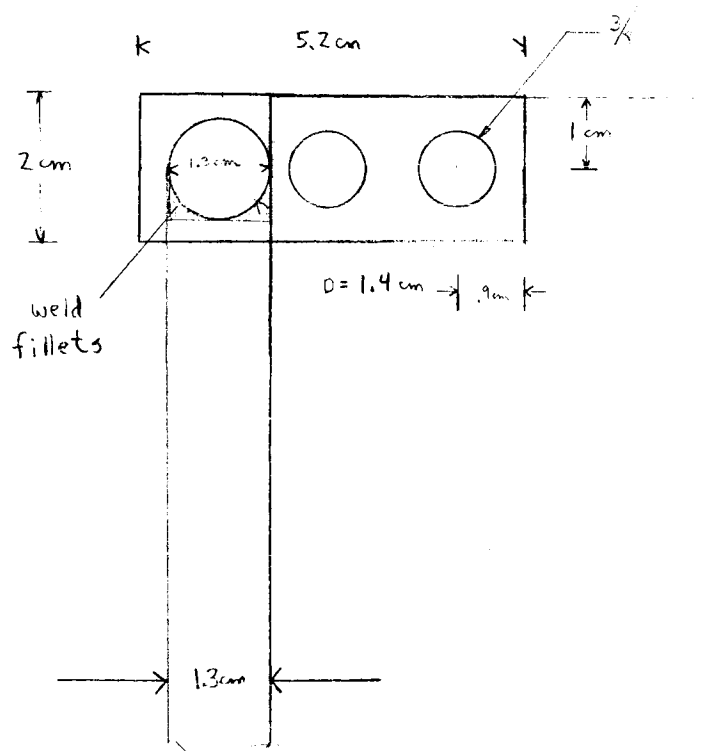
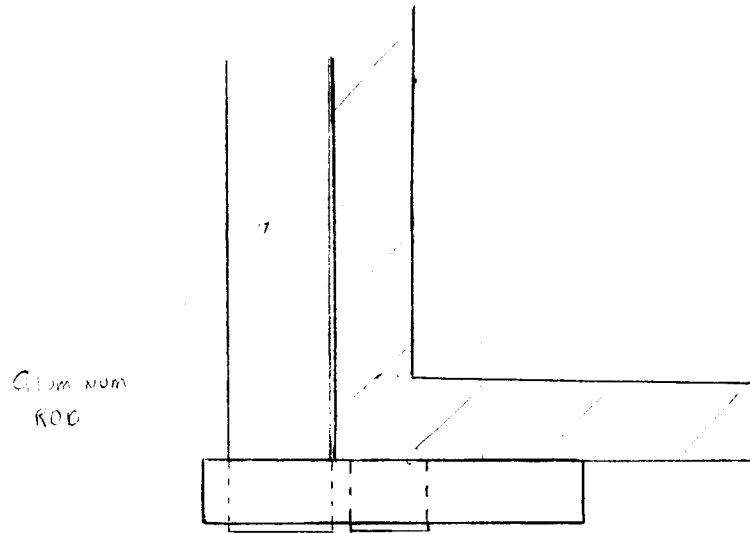
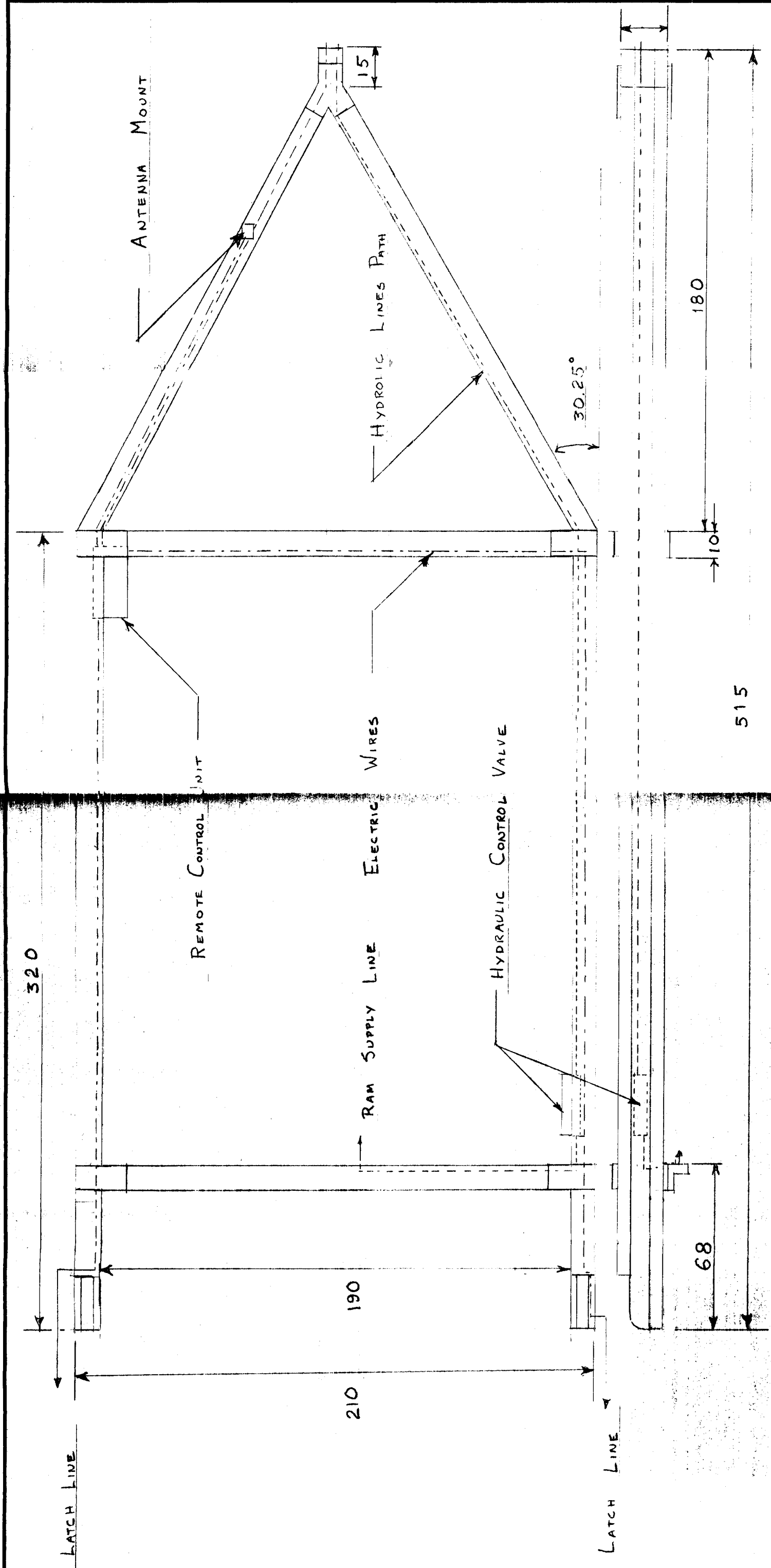


FIGURE 6

PIVOT POINT AT TOP OF DOOR:  
(full scale)

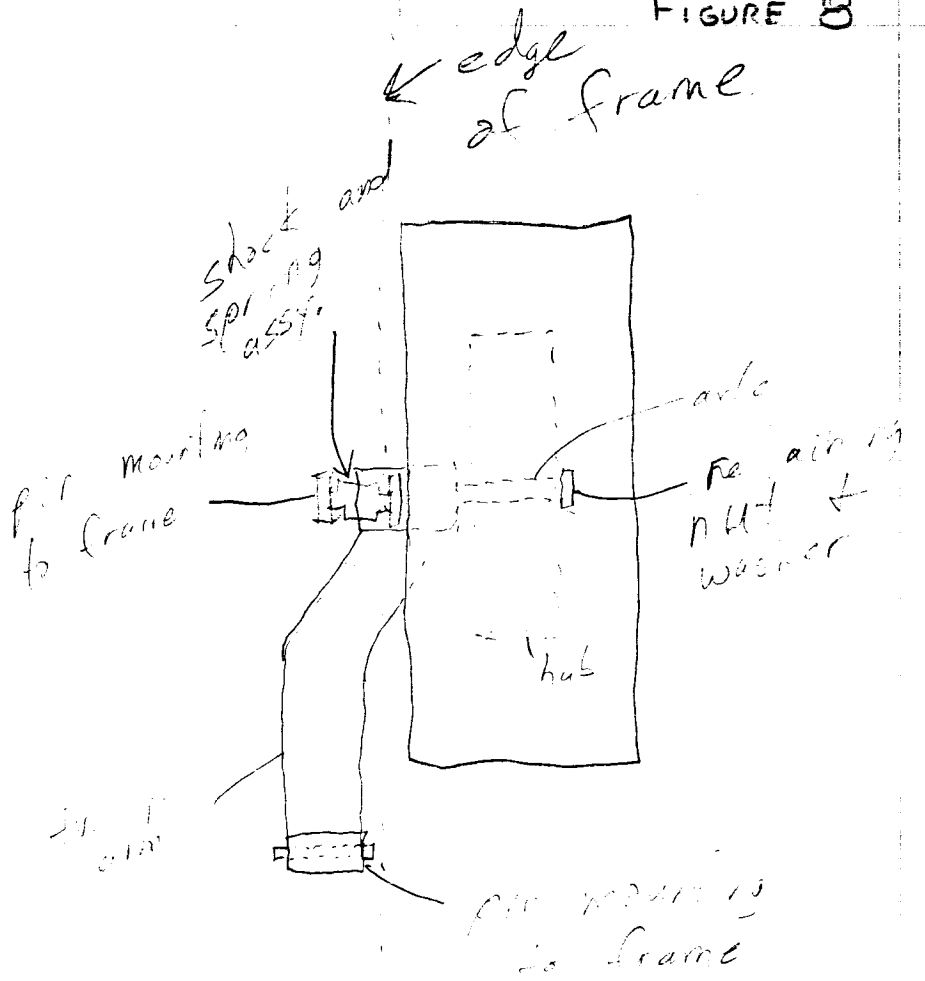




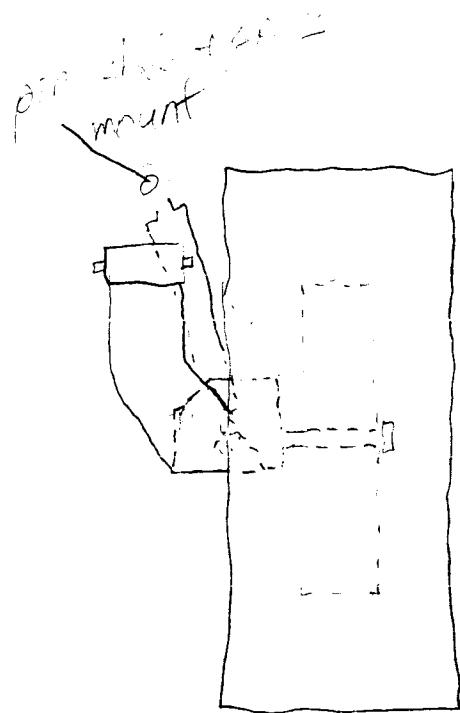
FRAME		APPROVED BY:	
SCALE: 1:40 CM		DRAWN BY A. BADE	
DATE: 1 JUNE 85		REVISED	
		DRAWING NUMBER	
		7	

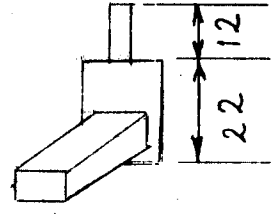
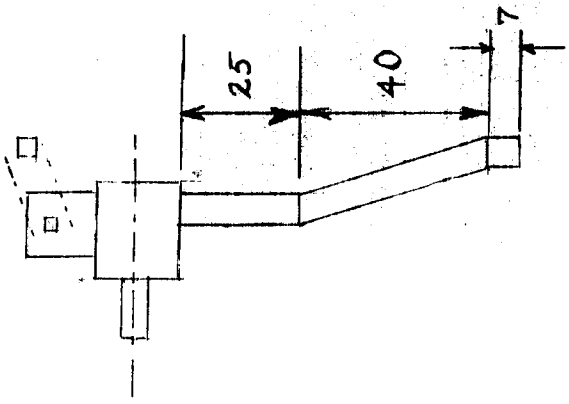
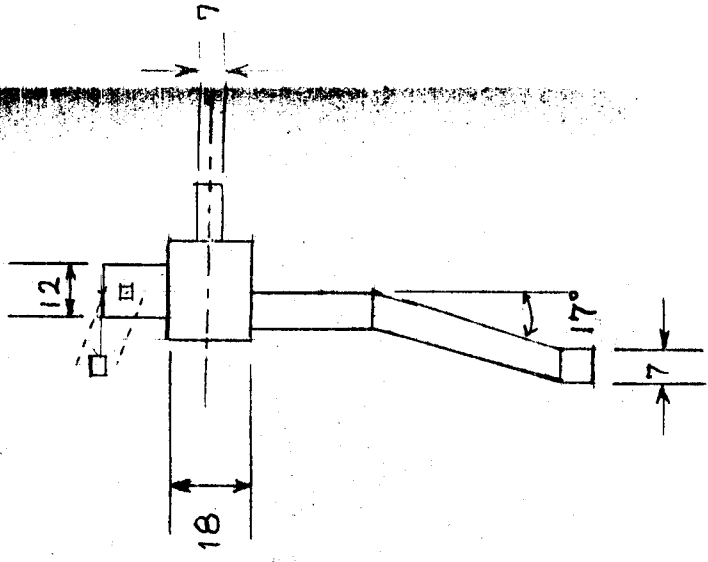
FIGURE 8

Top

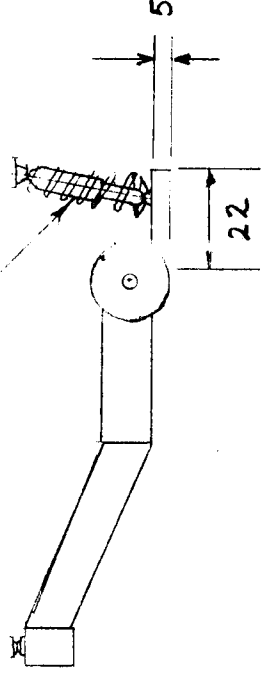


Front





COMMERCIAL SPRING & DAMPER



# REAR SUSPENSION

SCALE: 1:40 cm

DRAWN BY A. BADE

REVIS

DATE: 1 JUNE 85

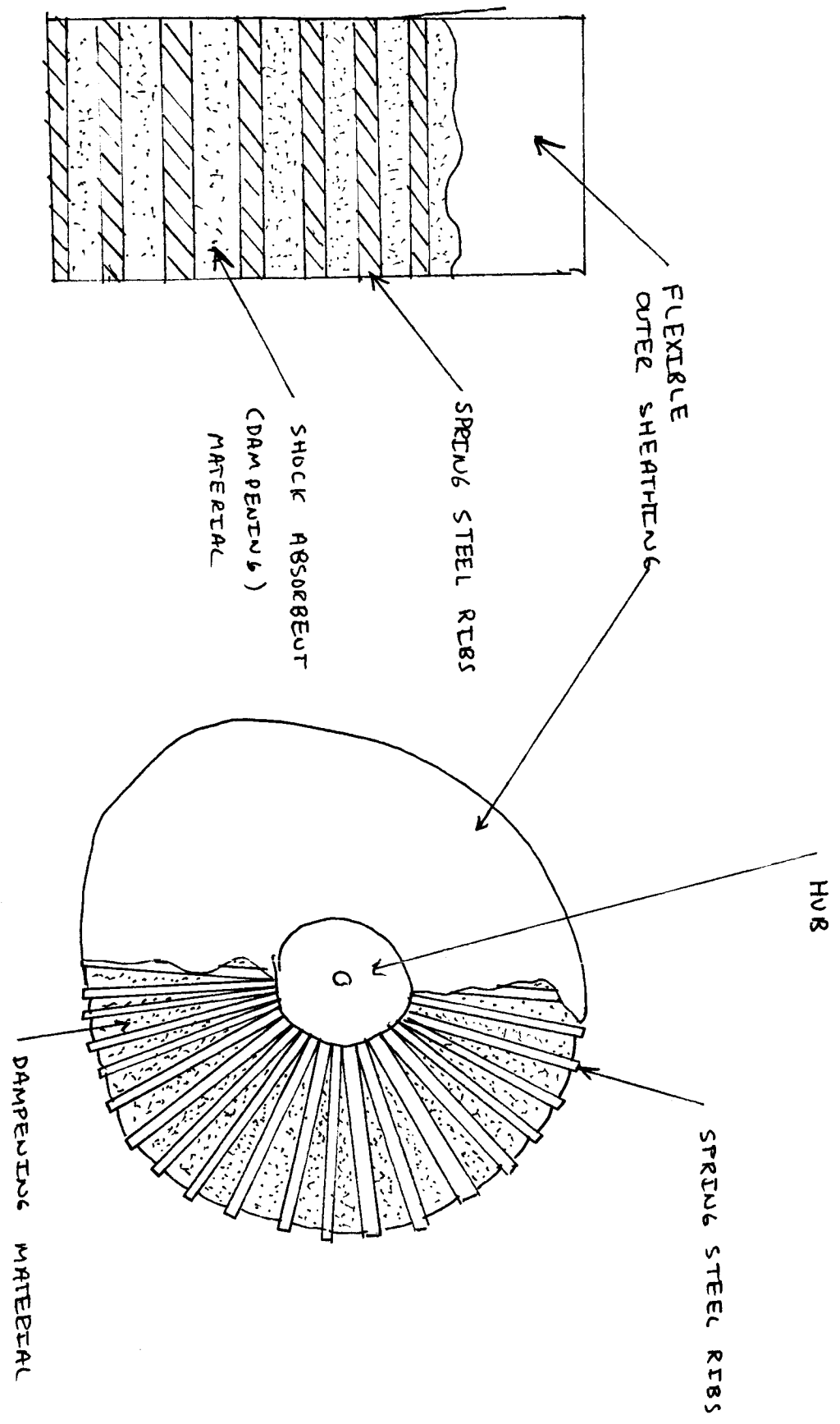
DRAWING NUMBER

9

WHEEL CONSTRUCTION

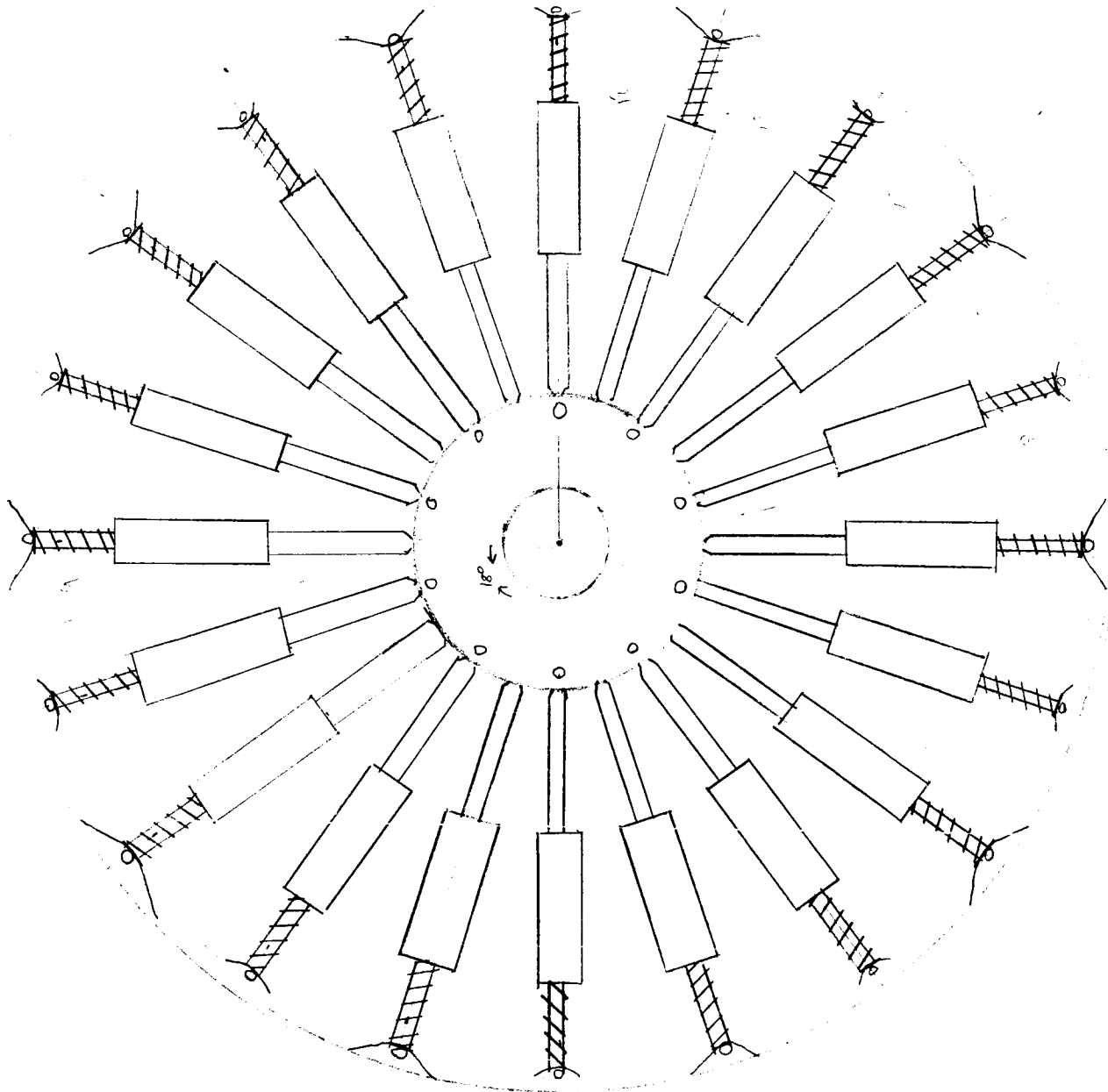
FIGURE 10

Design # 1

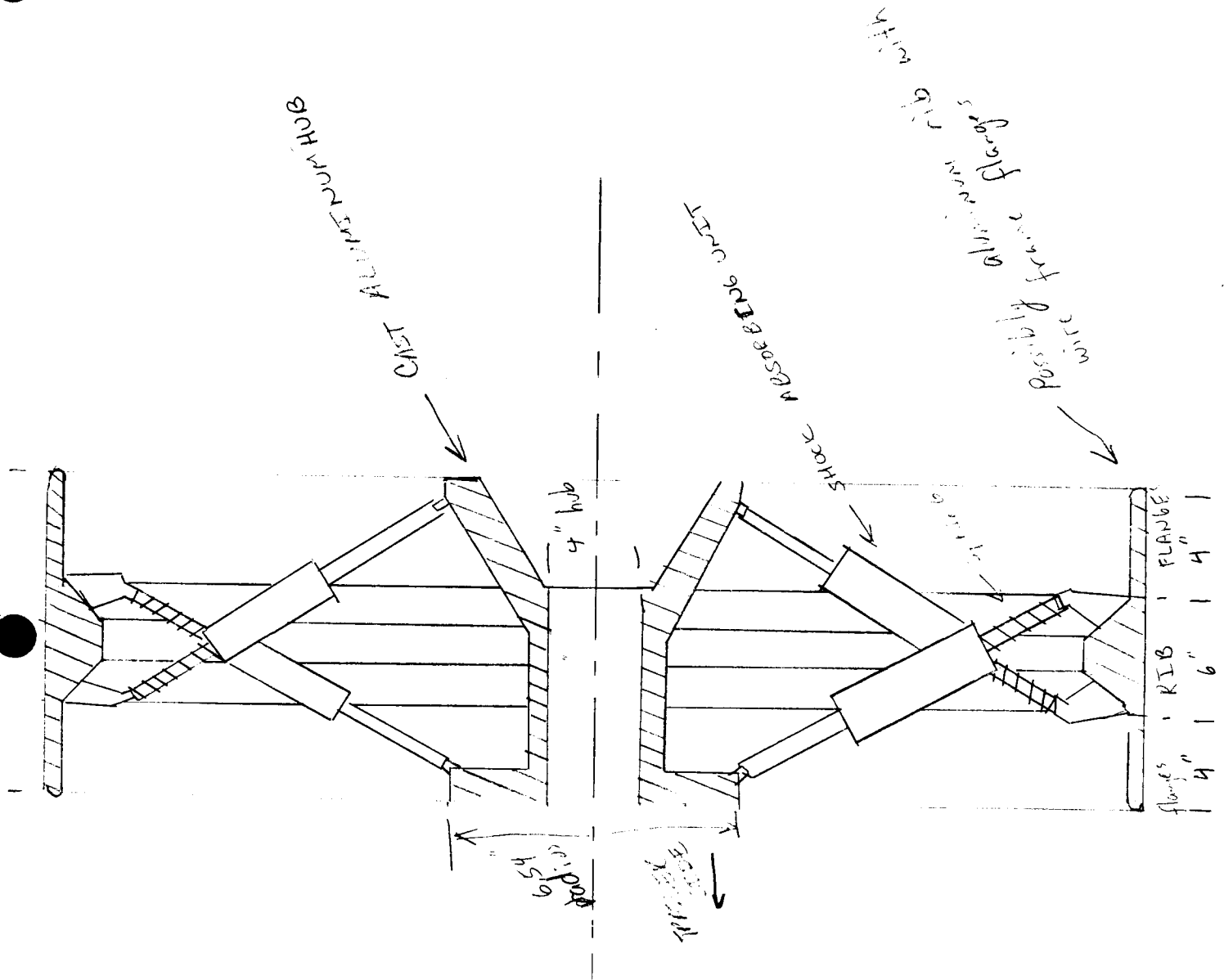


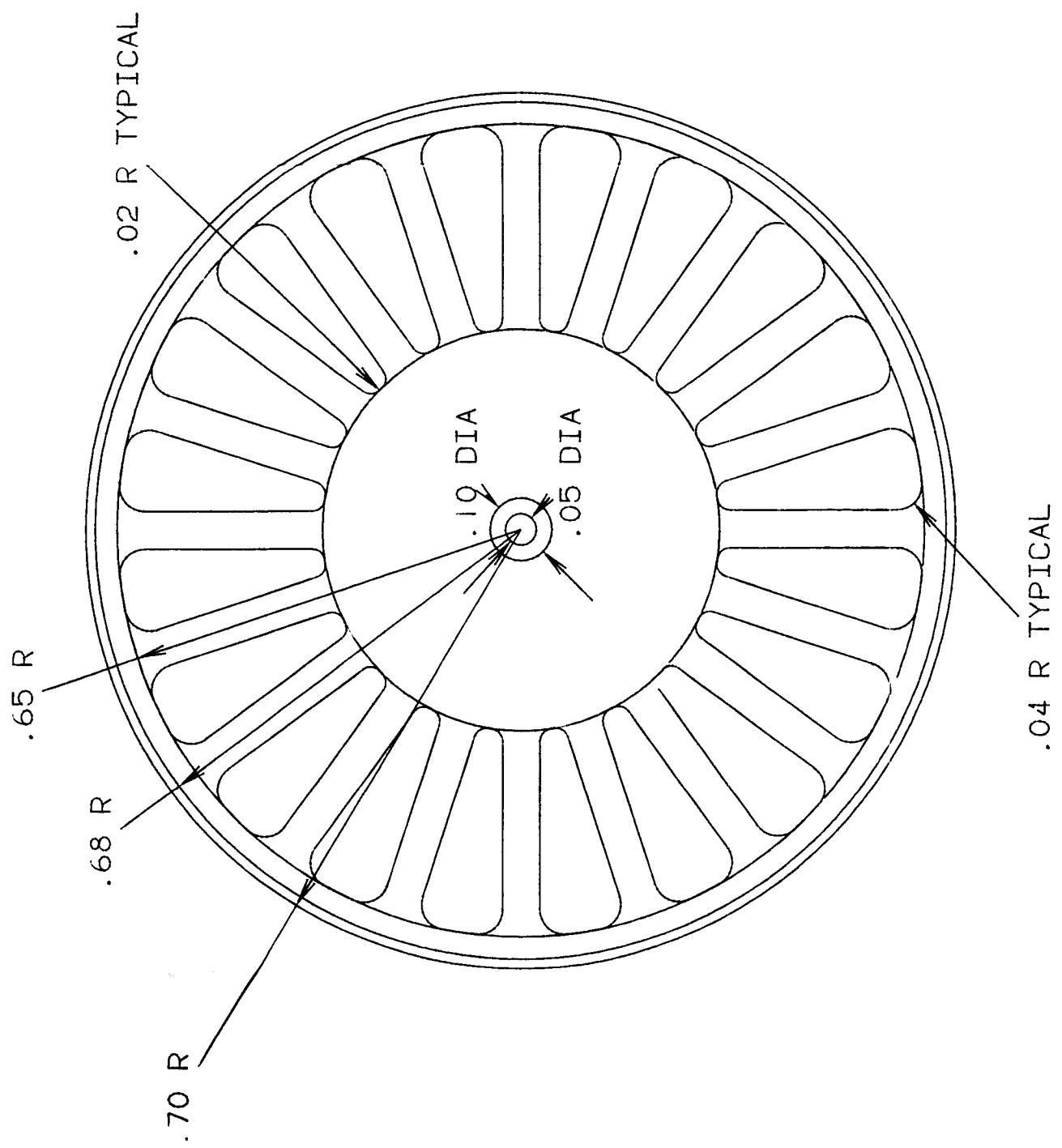
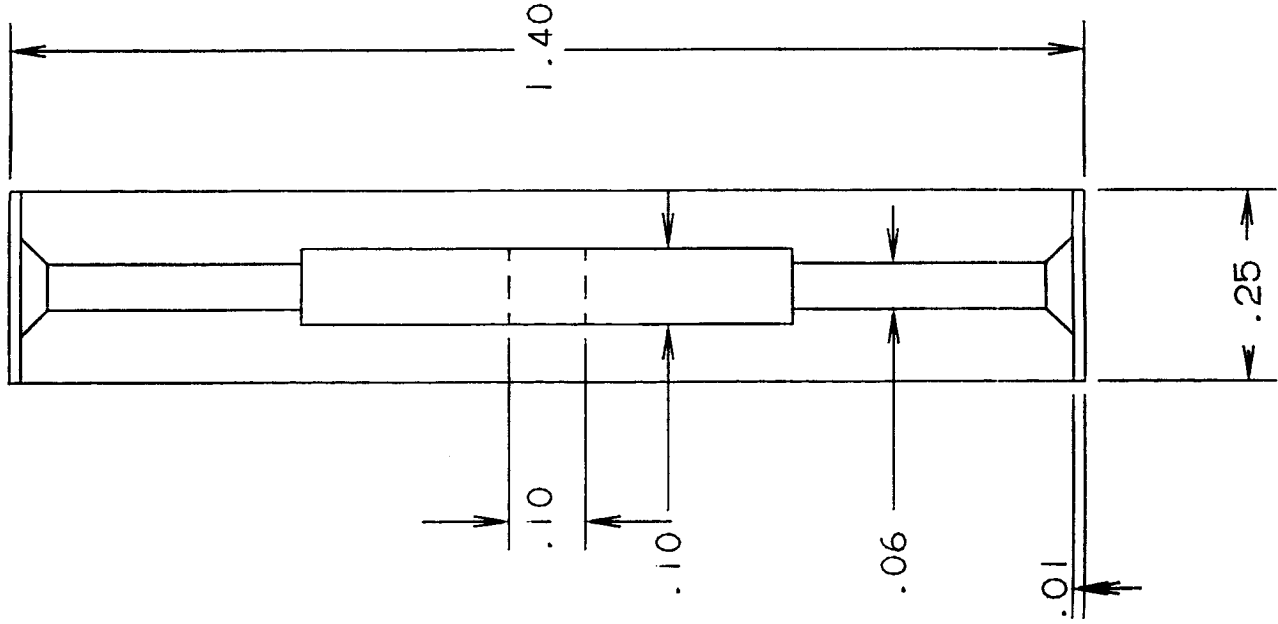
DESIGN #2

Hub radius - 6.5  
inner wheel - 2.5



DESIGN # 2

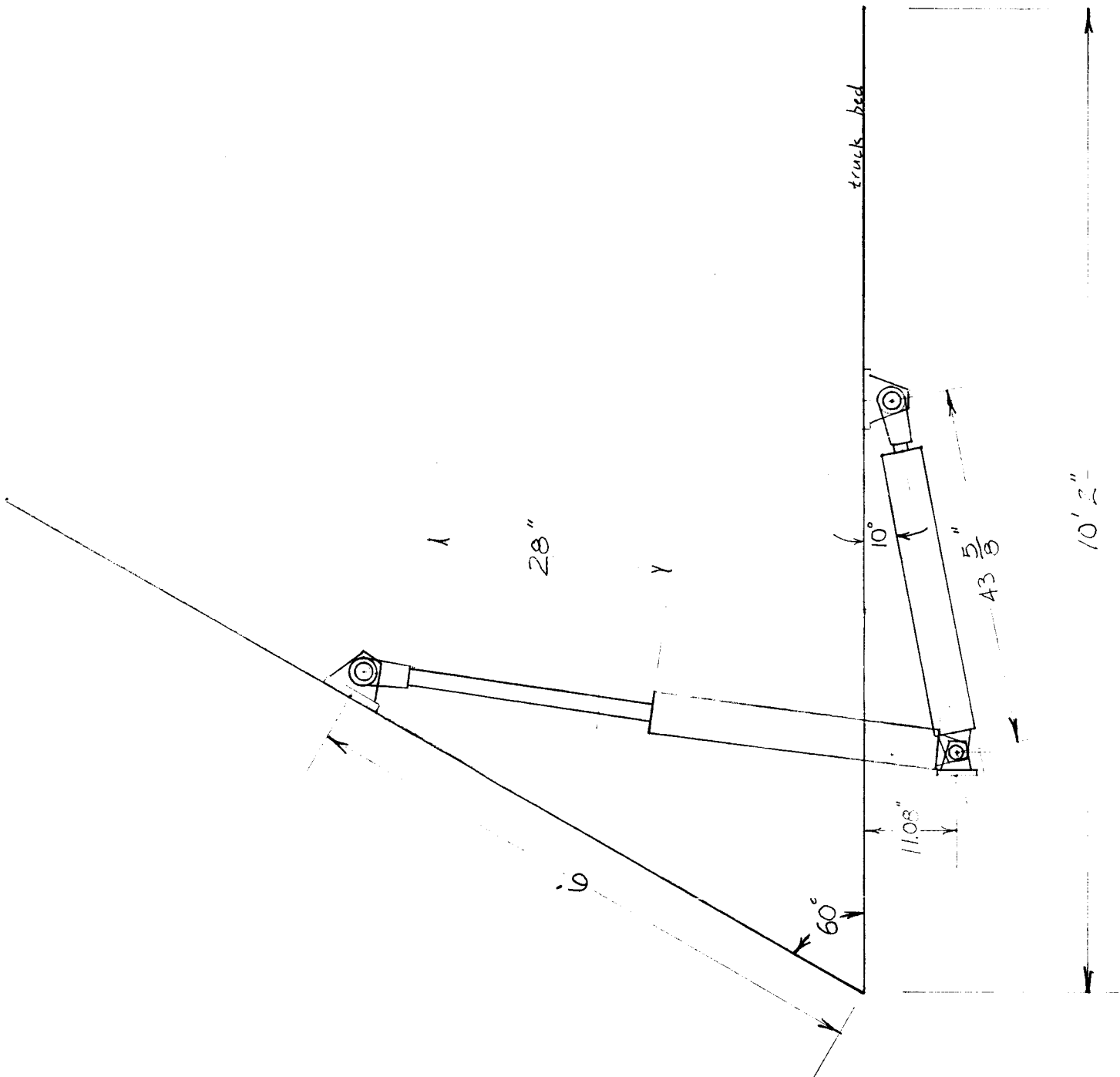




NASA / UNIVERSITY	
ADVANCED DESIGN CONCEPTS	
TITLE: LUNAR WHEEL DESIGN	
FORGED AL-95056-0	
DESIGN: A. BADE	DATE 6-3-85
CHECK:	DATE
DRWG NO. 12-A	

TOLERANCE UNLESS OTHERWISE NOTED:
IN: MM:
MATL:
SCALE:

FIGURE 13



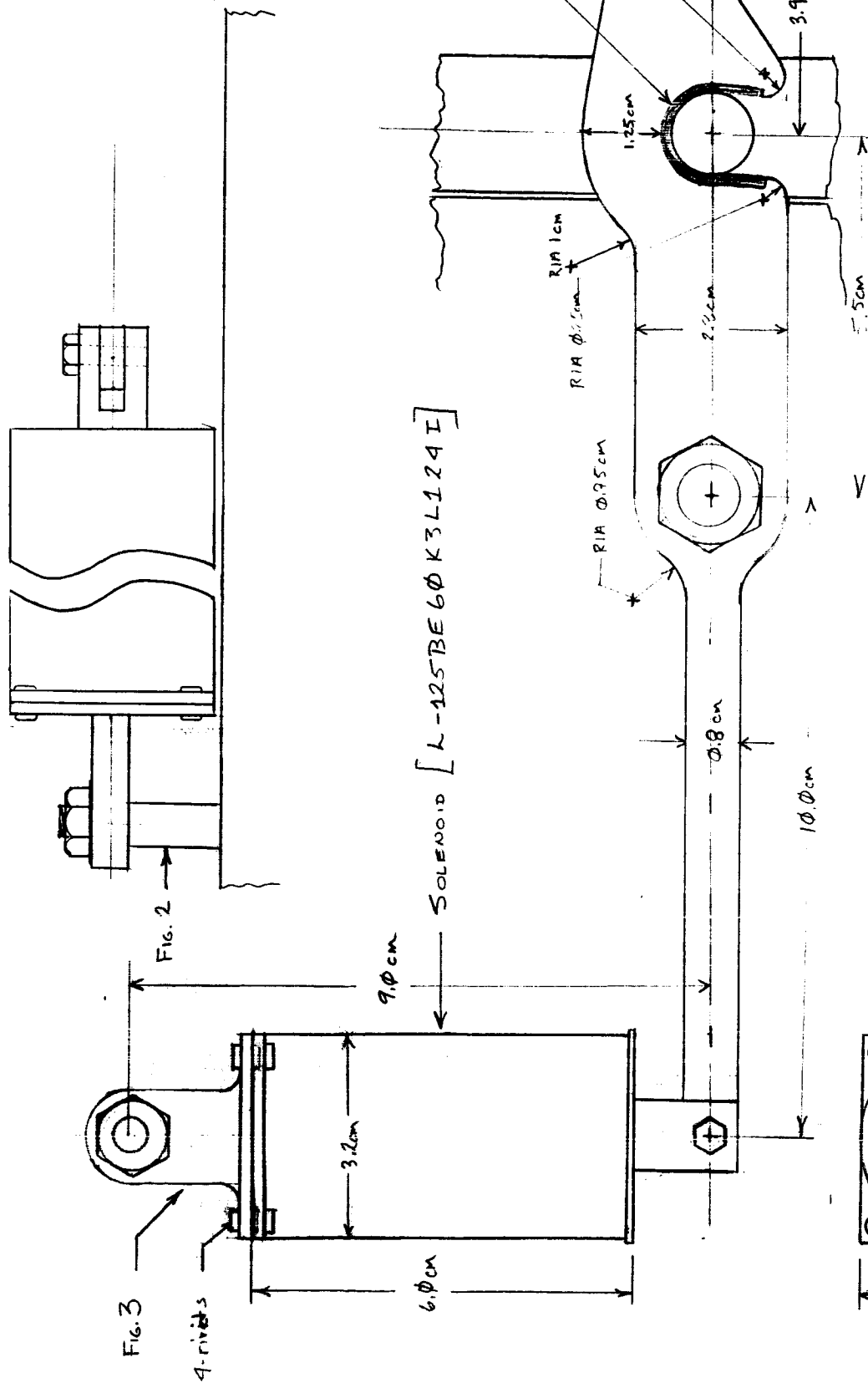


FIGURE 14

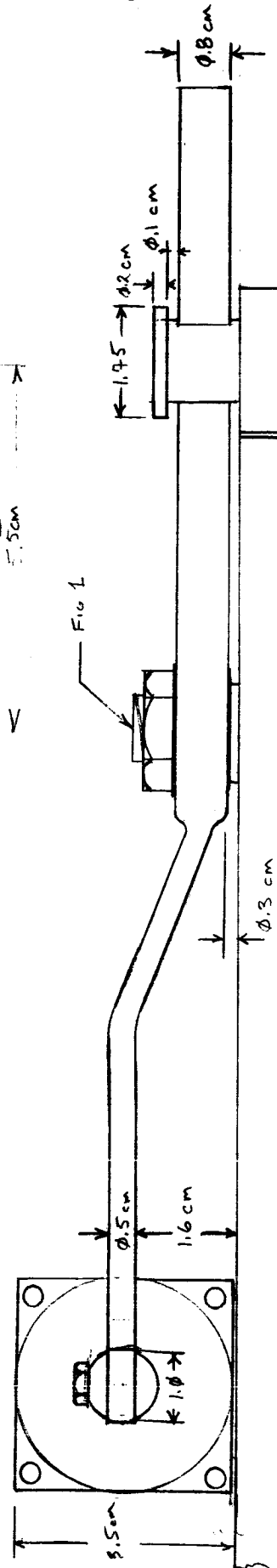


FIGURE A

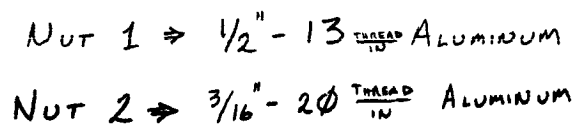
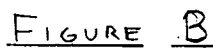


FIGURE 15

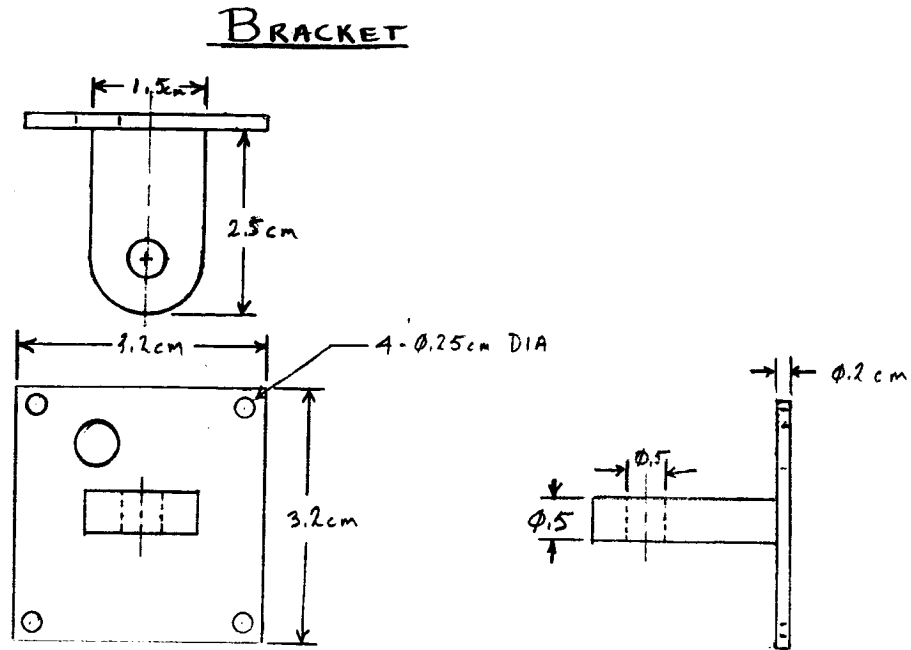


FIGURE 16

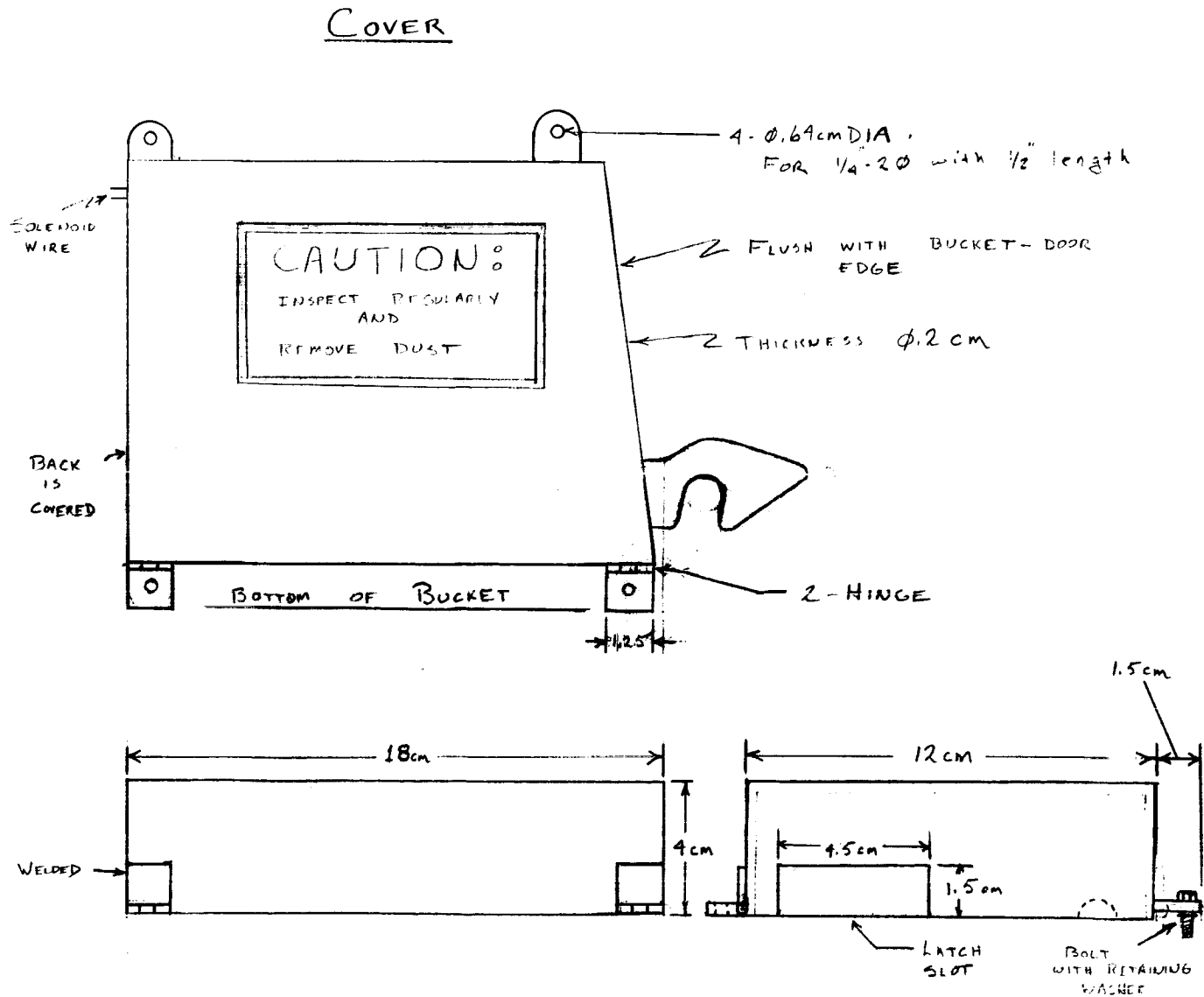
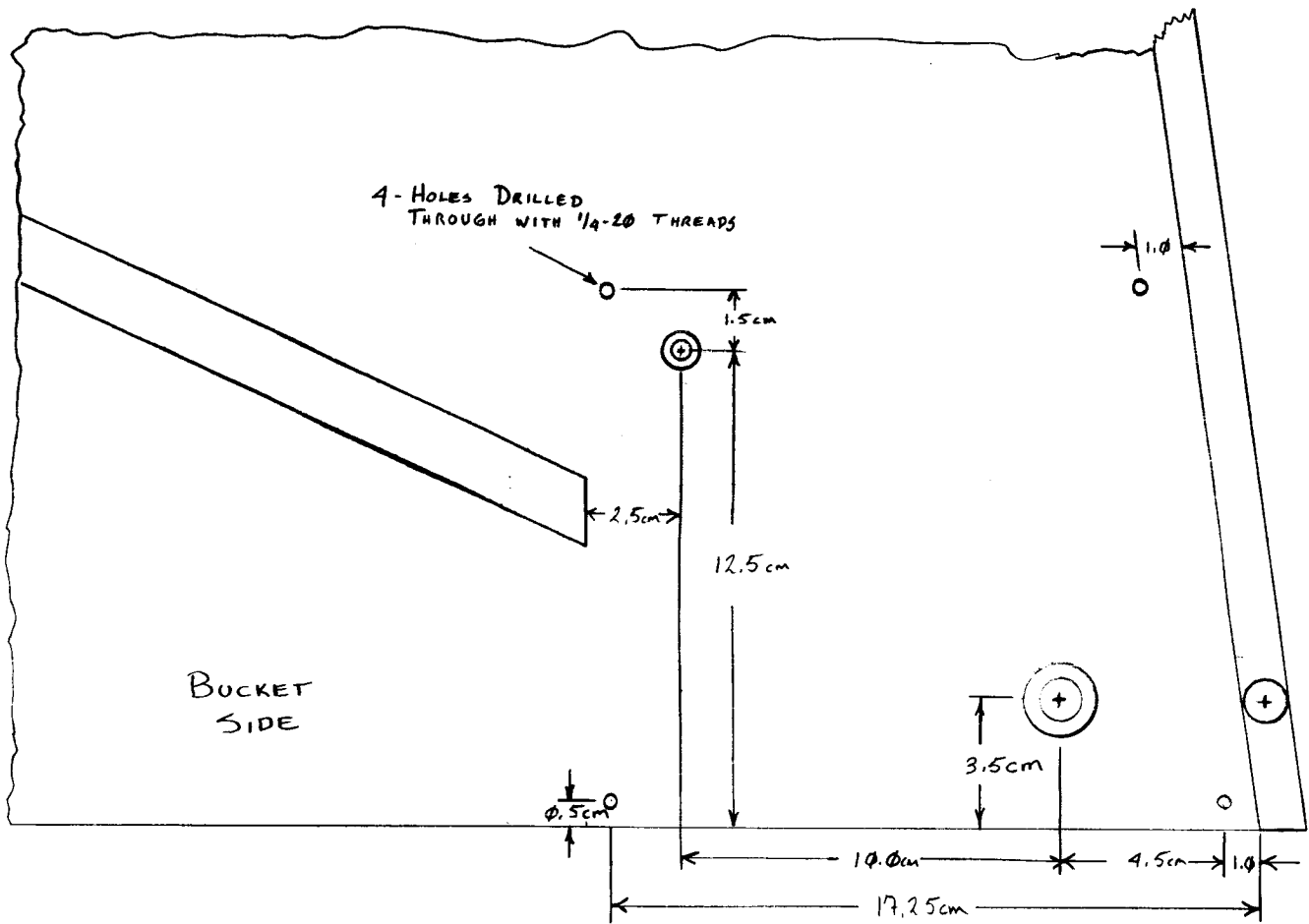


FIGURE 17



BUCKET ATTACHMENTS FOR  
DOOR LATCH



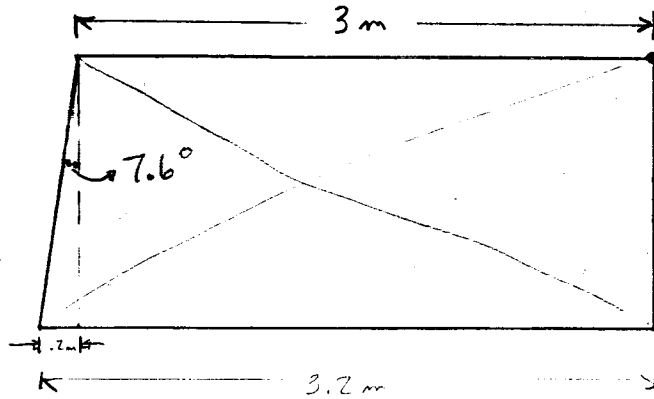
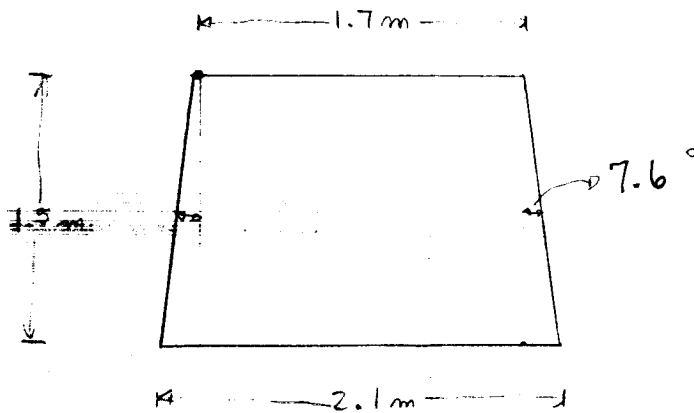
①

BEDside view:

$1'' = 1m$

Dimensions:

rear

front/rear view:Volume analysis:

$$\text{volume} = (\text{Area}_{\text{front}})(\text{length}) = \left[ \left( \frac{1}{2} \right) (1.7 + 2.1) (1.5) \right] [3.1]$$

$$= 8.84 \text{ m}^3$$

$$= 11.6 \text{ yd}^3$$

$$\frac{8.25}{x} = \frac{9.3}{12.2}$$

weight of soil:

moon earth density =  $1.7 \frac{\text{g}}{\text{cm}^3}$  (highest ave.)

$$1.7 \frac{\text{g}}{\text{cm}^3} \times \frac{1 \text{ kg}}{1000 \text{ g}} \times \frac{(100 \text{ cm})^3}{(1 \text{ m})^3} \times (8.84 \text{ m}^3) = 15,030 \text{ kg}$$

$$W_{\text{NET}} = mg = (15,030 \text{ kg}) \left( \frac{9.81}{(6)} \right) = 24,570 \text{ N} \text{ (highest weight of sand)}$$

In book, The Moon (by Harrison Gray) - Mariner

Temp. range on moon  $\Rightarrow -243^\circ\text{C}$  to  $30^\circ\text{C}$

FORCE ON WALLS:

Assume sand is roughly a fluid,

$$F_{\text{side, RESULTANT}} = \int w h dA$$

$$w = (1700 \frac{\text{kg}}{\text{m}^3}) \left( \frac{9.81 \text{ m/s}^2}{6} \right) = 2780 \frac{\text{N}}{\text{m}^3} = \frac{\text{N}}{\text{m}^3}$$

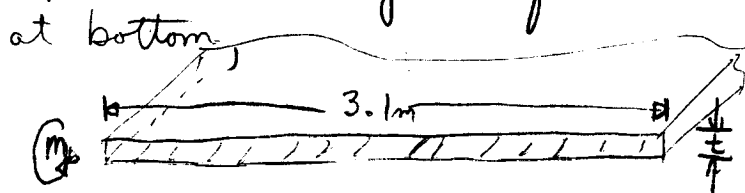
$$\begin{aligned} F_{s,R} &= w \int_0^{1.5} (1.5 - y)(3.1) dy = 8616 \left( 1.5y - \frac{1}{2}y^2 \right) \Big|_0^{1.5} \\ &= 8616 \left( 2.25 - \frac{1}{2}(2.25) \right) \\ &= 9693 \text{ N (moon force)} \end{aligned}$$

greatest moment at the bottom

$$M_{\text{bottom}} = (9693)(.75) = 7270 \text{ N}\cdot\text{m}$$

moment actually less.

Failure analysis for walls + bottom:



$$\begin{aligned} \sigma &= \frac{M c}{I} = \frac{M}{Z} & Z &= \frac{b h^2}{6} & b &= 3.1 \text{ m} \\ &= \frac{7260}{.517 t^2} & &= \frac{(3.1) t^2}{6} & h &= t \\ &= 1.404 \times 10^4 \frac{1}{t^2} & &= .517 t^2 \end{aligned}$$

$\sigma_y$  is given range of the  $\sigma$ :

at best, lightest and steepest slope are very strong + hard at low temps.

from Alcoa Handbook,

(al alloy 2024-T86) ( $\sigma_{y, \text{best}} = 67,000 \text{ psi}$ )

very hard at all temps (5%  $\sigma$  elongation)

$$67,000 \text{ psi} \times \frac{6900 \text{ N/m}^2}{1 \text{ psi}} = 4.623 \times 10^8 \text{ Pa} \left( \frac{\text{N}}{\text{m}^2} \right)$$

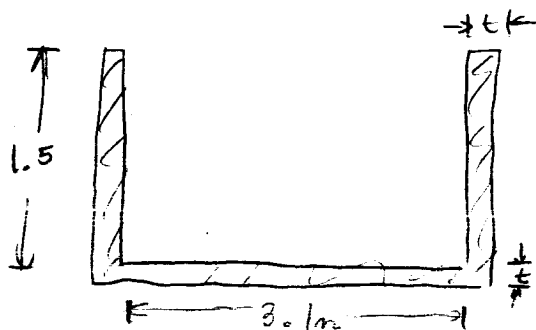
$$4.623 \times 10^8 = \frac{1.404 \times 10^4}{t^2}$$

$$t = 5.511 \times 10^{-3} \text{ m} = .551 \text{ cm}$$

safety factor of 2  $\rightarrow t = 1.102 \text{ cm}$

4

axis



test normal stress failure

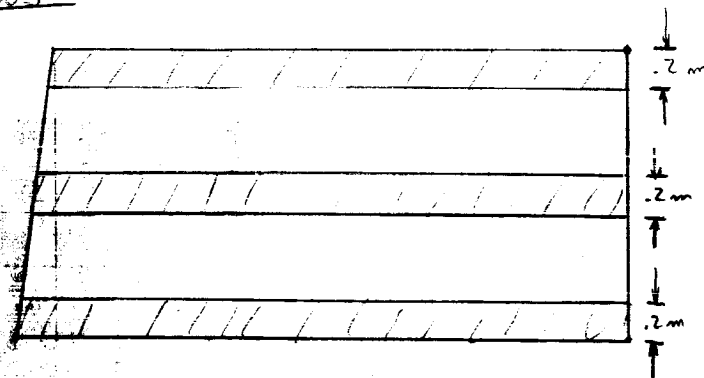
$$\text{total x-sectional area} = (3.0)(.011) + (2)(1.5)(.011) + 2(.011)^2$$

$$= .06734 \text{ m}^2$$

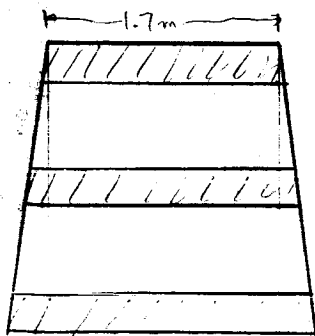
$$\frac{F_R}{A_{x\text{-sec}}} = \frac{9700}{.06734} = 144,040 \text{ Pa (well below } \sigma_y)$$

FIRST ATTEMPT AT RIBS:  
for added safety, add ribs. (2x the required t)

side view:

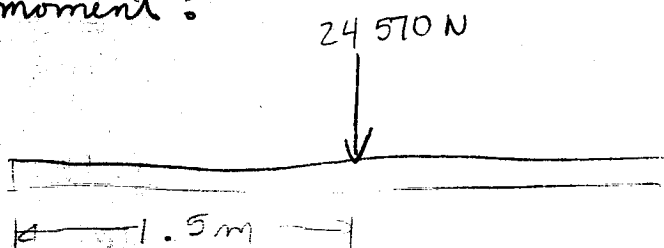


rear + front:



(5)

bottom moment:



$$(24570 \text{ N})(1.5 \text{ m}) = 36,900 \text{ N}\cdot\text{m}$$

Weight of bed analysis:weight of bed:

$$V_{\text{total}} = 2 V_{\text{front/side}} + V_{\text{bottom}}$$

$$V_{\text{FRONT}} = \left[ \frac{1}{2} (2.1 + 1.7)(1.5) \right] [.011] + [.011] \left[ \frac{1}{2} (1.9 + 2.0)(.2) \right]$$

$$= .03135 + .01287 = .04422 \text{ m}^3$$

$$V_{\text{side}} = \left[ \frac{1}{2} (3 + 3.2)(1.5) \right] [.011] + [.011] \left[ \frac{1}{2} (3.1 + 2.5)(.2) \right]$$

$$= .05115 + .02046 = .07161 \text{ m}^3$$

$$V_{\text{bottom}} = (3.2)(2.1)(.022) = .14784 \text{ m}^3$$

$$V_t = .3795 \text{ m}^3$$

6

p. 74 - Alcoa Handbook

for chosen alloy,

for flat sheet,

$$\sigma_y = 58,000 \text{ psi} = 400 \times 10^6 \text{ N/m}^2$$

p. 91:

$$\text{sample piece} - 36" \times 96" = 3456 \text{ in}^2 \times \frac{(0.3048 \text{ m})^2}{144 \text{ in}^2} = 2.23 \text{ m}^2$$

$$\text{area of base} = 6.72 \text{ m} \rightarrow 24,510 \text{ N}$$

$$2.23 \text{ m} \rightarrow 8153 \text{ N} \times \frac{1 \text{ lb}}{4.45 \text{ N}} = 183 \text{ lb}$$

use safety factor of 4 for bottom for fallings rocks:

$$t_b = 4 \times .00551 = .022 \text{ m} = 2.2 \text{ cm}$$

$$\cancel{a1 \ 2024} = \frac{346 \cancel{\text{ lb}}}{2.25 \cancel{\text{ m}^3}} \times \frac{4.45 \cancel{\text{ N}}}{1 \cancel{\text{ lb}}} \times .3795 \cancel{\text{ m}^3} = 2.77 \times 10^4 \frac{\text{N}}{\text{m}^3}$$

$$a1 \ 2024 = \frac{.4 \text{ lb}}{\text{in}^3} \times \frac{(0.0254 \text{ m})^3}{(0.0254)^3 \text{ m}^3} \times \frac{4.45 \text{ N}}{1 \text{ lb}} = 2.72 \times 10^4 \frac{\text{N}}{\text{m}^3}$$

$$\text{weight} = w \times V = (2.72 \times 10^4)(.3795) = 10,300 \text{ N}_{\text{earth}} \\ = 1717 \text{ N}_{\text{moon}}$$

7

$$\text{total moon weight of bed + load} = \boxed{\begin{matrix} 26,300 \text{ N} \\ 5900 \text{ lb} \end{matrix}}$$

Heat up in wheels:

Heat transfer in wheels:

$$\text{radius of wheel} = 2 \text{ ft} \times \frac{.3048 \text{ m}}{1 \text{ ft}} = .61 \text{ m}$$

at equil.,  $Q_{in} = Q_{out}$

$$A_{x-sfc} = \pi (.1)^2 = .03142 \text{ m}^2$$

$q_{in} = 25 \text{ hp}$  ← pulling power

$$k_{Al} = 236 \frac{\text{W}}{\text{m K}}$$

2 wheels → 12.5 hp/wheel

$$q_{in} = \epsilon \sigma A (T_w^4 - T_s^4) + k A_x (T_w - T_{sfc})$$

$$q_{in} = 12.5 \text{ hp}, \frac{745.7 \text{ W}}{1 \text{ hp}} = 9321 \text{ W}$$

$\epsilon = 1.0$  (black bodies)

$$9321 = \epsilon \sigma A (T_w^4 - T_s^4) + k A_x (T_w - T_{sp}) \quad \approx 30^\circ \text{K}$$

$$= (5.67 \times 10^{-8}) (\pi (.61)^2) (T_w^4 - T_s^4) + (236) (.03142) (T_w - T_b)$$

$$9321 = (6.63 \times 10^{-8}) (T_w^4 - 810,000) + (7.42) (T_w - 30)$$

$$T_w^4 + 111,915.5354 T_w - 9543.6537 = 0$$

$$T_w = 0.0001$$

$$\textcircled{2} -482$$

$$\textcircled{3} \text{ im}$$

$$\textcircled{4} \text{ im}$$

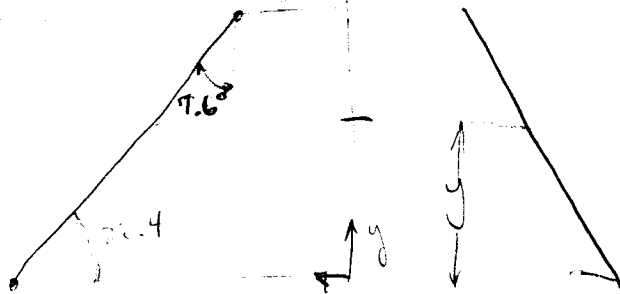
8

FIND FULL MARK:

$$10 \text{ yd}^3 \times \frac{8.84 \text{ m}^3}{11.6 \text{ yd}^3} = 7.62 \text{ m}^3$$

needed

$$V = 10 = \frac{1}{2} (3.1 + x) y$$



$$y = 7.5x + b$$

at  $x = 1.05, y = 0$

$$0 = -7.875 + b$$

$$b = +7.875$$

$$y = -7.5x + 7.875$$

$$x = \frac{-y + 7.875}{7.5}$$

$$V = \left[ \frac{1}{2} (1.05 + x) (y) \right] \times [3.1] = 3.81$$

$$\frac{1}{2} \left( 1.05 + \left( \frac{-y + 7.875}{7.5} \right) \right) (y) (3.1) = 3.81$$

$$1.05y - 1.333y^2 + 1.05y = 2.4581$$

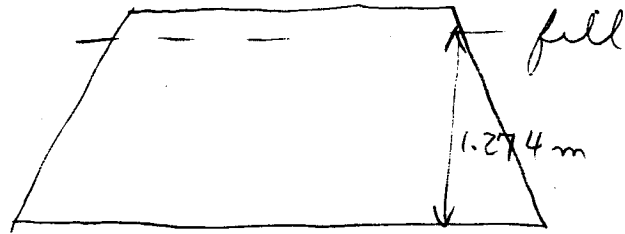
$$y^2 - 15.75y + 18.4358 = 0$$

$$y = 14.47 \text{ or } 1.2735 \text{ m}$$

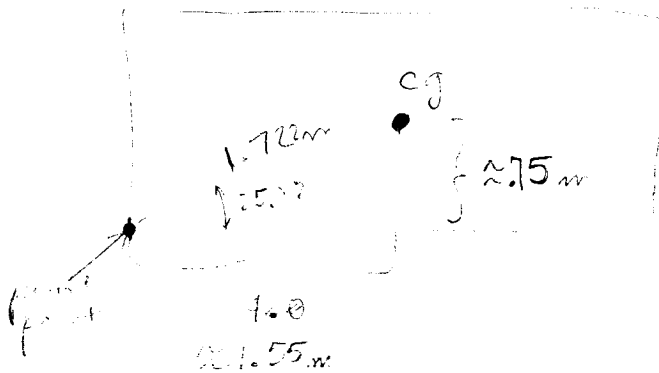
just below top sill

9

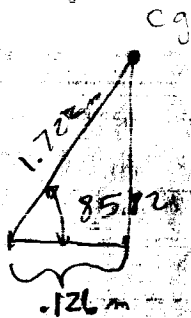
Location of cg at certain times :

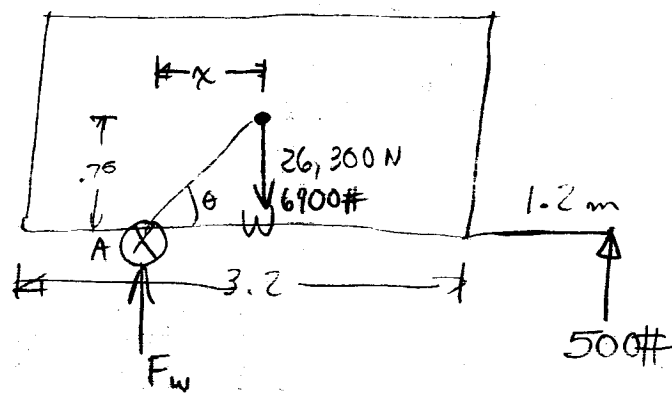


$$\frac{1}{2}(2.1 + x) y =$$



after rotation :





$$M_a = \sum \tau = (6900)x - (500)(3.2)$$

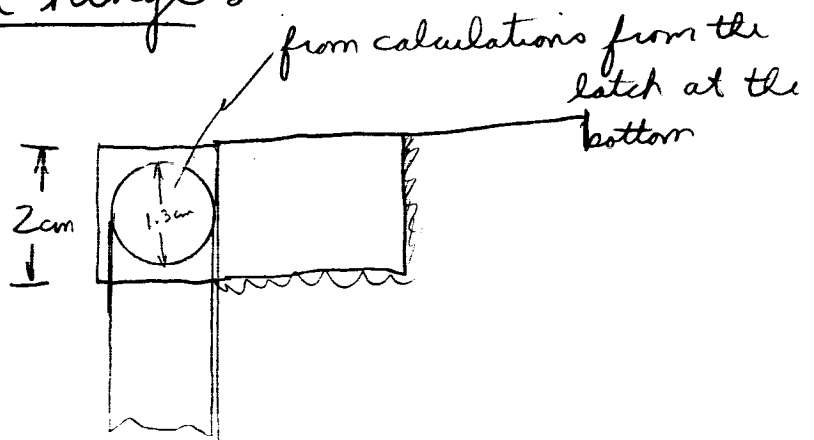
$$x = .2$$

$$\theta = \tan^{-1}\left(\frac{0.75}{0.2}\right) = 75^\circ$$

# Design of upper hinge:

400N

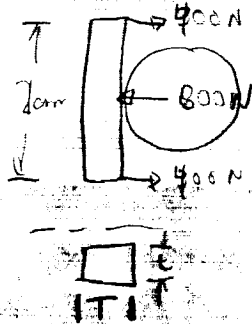
$$\frac{400}{(.006)(.007)}$$



force need to hold lid against back =  $\frac{1}{2}(1600) = 800N$

↑  
pressure resultant  
down due to sand  
pressing outward

worst case for failure:



$$M_{at\ middle} = (62)(300)(1) = 18600 N \cdot m$$

$$t = .008 m =$$

↑  
from calculations from latch again

$$\tau_y = \frac{M}{Z}$$

$$Z = \frac{M}{\tau_y} = \frac{18600}{4.62 \times 10^8} = 1.73 \times 10^{-8}$$

$$Z = \frac{bh^3}{6} = \frac{(.008)h^3}{6} = 1.3 \times 10^{-8}$$

$$h = .00362 = .36 cm \Rightarrow \times 2 = .6 cm$$

safety factor  $\frac{1}{2}$

bolts holding on bracket

$$\tau_y = \tau_y = \frac{m}{z} \quad m = \frac{Ft}{2}$$

$$F = \frac{600}{2 \text{ bolts}} = 300 \frac{\#}{\text{bolt}}$$

$$m = \frac{(300)(.008)}{2} \leftarrow t \text{ of bracket} = 1.2 \text{ N}\cdot\text{m}$$

$$\tau_{a1} = 4.62 \times 10^8 = \frac{1.2}{z}$$

$$z = 2.6 \times 10^{-9} = \frac{\pi d^3}{32}$$

$$d = .003 \text{ m} = .3 \text{ m} \times \frac{141}{.3048 \text{ m}} \times \frac{12''}{141} = .117''$$

use  $\frac{1}{4}''$  hexagon head cap screws (ASA B18.2)

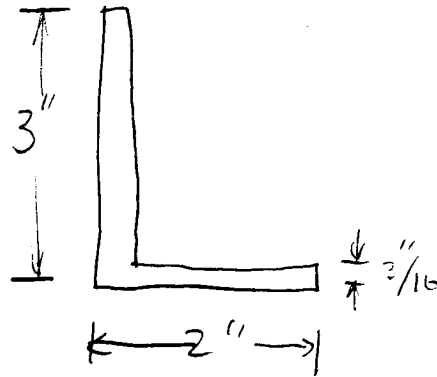
↑  
3181 ksi (< 114 ksi)

Design of rebs:

$$I = 2.75 \times 10^{-6} \text{ m}^4 \times \frac{(1 \text{ ft})^4}{(.3048 \text{ m})^4} \times \frac{(12 \text{ in}^4)}{(1 \text{ ft})^4}$$

$$I_{\text{new}} = \frac{(3.1)(.011)^3}{12} = 3.44 \times 10^{-7} \text{ m}^4 = .826 \text{ in}^4$$

use standard unequal angle L beam

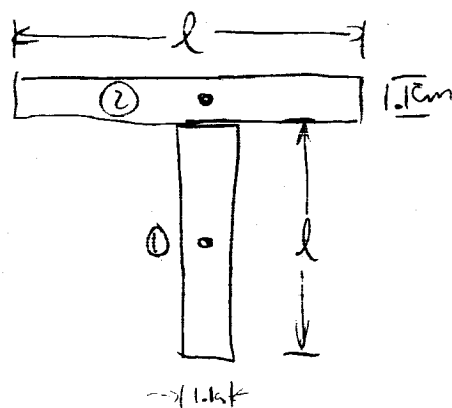


safety factor of walls = 6

$$2" = \times \frac{12 \text{ in}}{\text{ft}} \times \frac{1 \text{ ft}}{.3048 \text{ m}} = .051 \text{ m} = 5.1 \text{ cm}$$

$$3" =$$

T-beam analysis:  
using T-beam,



we want  $I$  to add safety factor of  $[4]$  more

$$I = \frac{bh^3}{12} = \frac{(3.2)(0.011)^3}{12} = 2(3.55 \times 10^{-7}) m^4 = 7.1 \times 10^{-7}$$

$$I_1 = \frac{(0.011)(l^3)}{12} + (0.011)(l)\left(\frac{l}{2}\right)^2 = I_v + A d^2$$

$$I_2 = \frac{(l)(0.011)^3}{12} + (0.011)(l)(l^2)$$

$$I_{T+} = .000917 l^3 + .00275 l^2 + 1.11 \times 10^{-7} l + .04 l^3 = 7.1 \times 10^{-7}$$

$$.012 l^3 + .00275 l^2 + 1.11 \times 10^{-7} l - 7.1 \times 10^{-7} = 0$$

$$l^3 + .23 l^2 + 9.25 \times 10^{-6} l - 5.91 \times 10^{-5} = 0$$

$$l = .0156, -.0117, -.2294$$

Round bar for bed:

$$\tau = .577 S_y = 266 \times 10^6 \text{ Pa}$$

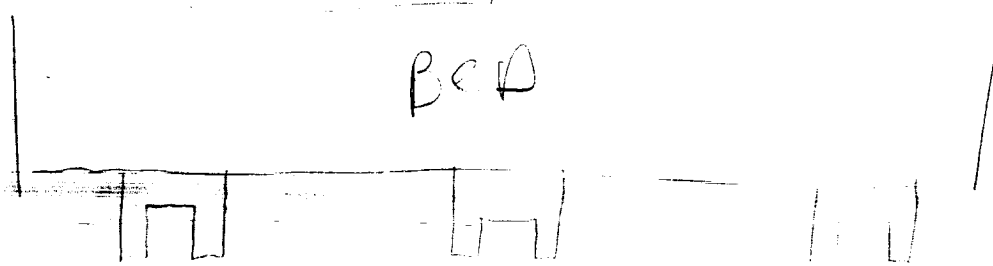
$$\tau = \frac{F}{A}$$

$$A = \frac{F}{\tau} = \frac{30000}{266 \times 10^6} = 1.13 \times 10^{-4} \text{ m} = \pi R^2$$

$$R = .006 \text{ m}$$

safety factor 7  $\rightarrow R = .042 \text{ m}$   
 $= 4.2 \text{ cm}$   
 $\approx 4 \text{ cm}$

$$D = 8 \text{ cm}$$



weight of bed:

$$V_t = V_B + 2 V_{side} + 2 V_{fb} + V_{RIBS} + V_{etc}$$

$$V_B = (3.2)(2.1)(.022) = .148 \text{ m}^3$$

$$V_{fb} = \left[ \frac{1}{2} (2.1 + 1.7) \right] [1.5] \left[ \frac{.011}{.274} \right] = \cancel{.063} .0314$$

$$V_{sm} = \left[ \frac{1}{2} (3.2 + 2.0) \right] [1.5] \left[ \frac{.011}{.107} \right] = \cancel{.107} .0511$$

$$V_{etc} = (1.6) \left[ \left( \frac{.010}{.01} \right) \right] \left[ (4)(3.53) + 4(2.5) \right]$$

$$= \cancel{.0075} .0025$$

$$V_t = \cancel{.56} \text{ m}^3 = .4 \text{ m}^3$$

$$W_{etc} = \cancel{(.56)} (2.45 \times 10^4 \text{ N}) =$$

# FRAME

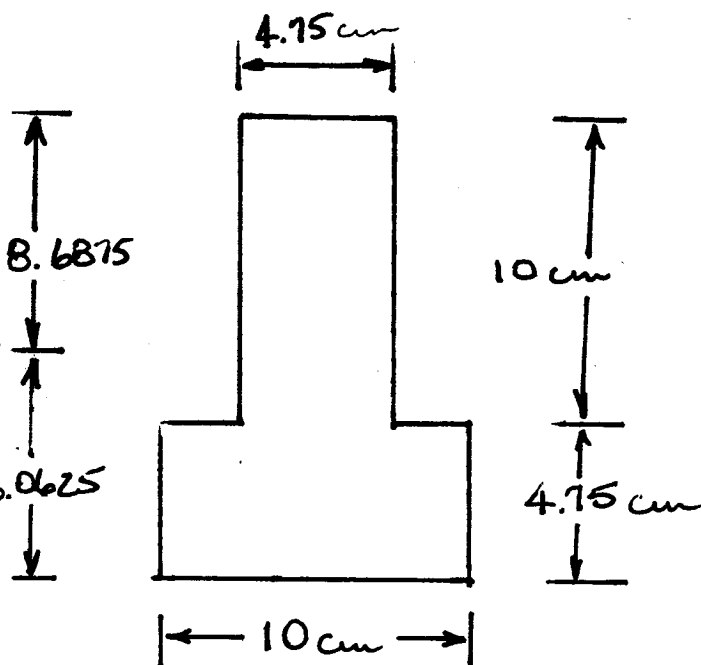
## T BEAM ANALYSIS

MATERIAL:

$$\text{AL2024-T86}$$
$$G_y = 4.8 \times 10^8 \text{ Pa}$$

$$C_2 = 8.6875$$

$$C_1 = 6.0625$$



ANALYSIS TO ENSURE STRENGTH REQUIRED TO SUPPORT LOADED BUCKET. WORSE CASE USED - ENTIRE LOAD ON SINGLE WEBB.

$$A = 2(10 \text{ cm})(4.75 \text{ cm}) = 95 \text{ cm}^2$$

$$95 C_1 = (4.75)(10)\left(\frac{4.75}{2}\right) + (4.75)(10)(9.75)$$

$$\Rightarrow C_1 = 6.0625 \text{ cm}$$

$$C_2 = 14.75 \text{ cm} - 6.0625 \text{ cm} = 8.6875 \text{ cm}$$

$$I_1 = \frac{b h^3}{12} = \frac{(10)(4.75)^3}{12} = 89.31 \text{ cm}^4$$

$$I_2 = \frac{(4.75)(10)^3}{12} = 395.83 \text{ cm}^4$$

$$d_1 = C_1 - \frac{4.75}{2} = 6.0625 - \frac{4.75}{2} = 3.6875 \text{ cm}$$

$$d_2 = C_2 - 5 = 8.6875 - 5 = 3.6875 \text{ cm}$$

$$I_T = (I_1 + A_1 d_1^2) + (I_2 + A_2 d_2^2) = I_1 + I_2 + 2 A d^2$$

$$= 89.31 + 395.83 + 2(4.75)(10)(3.6875)^2$$

$$I_T = 1776.92 \text{ cm}^4$$

# FRAME

$$\sigma = \frac{Mc_1}{I} = \frac{(0.65m)(27000N)(0.060625m)}{1776.92 \text{ cm}^4 \left(\frac{1m}{100 \text{ cm}}\right)^4}$$

$$\sigma = 6.0 \times 10^7 \text{ Pa}$$

$$\sigma_y \text{ of ALUMINUM} = 4.8 \times 10^8 \text{ Pa}$$

$$n = \frac{4.8 \times 10^8}{6.0 \times 10^7} = 8$$

THEREFORE, IN WORST CASE CONDITIONS, THE BEAM IS STRONG ENOUGH.

# FRAME

## FRAME BOLTS

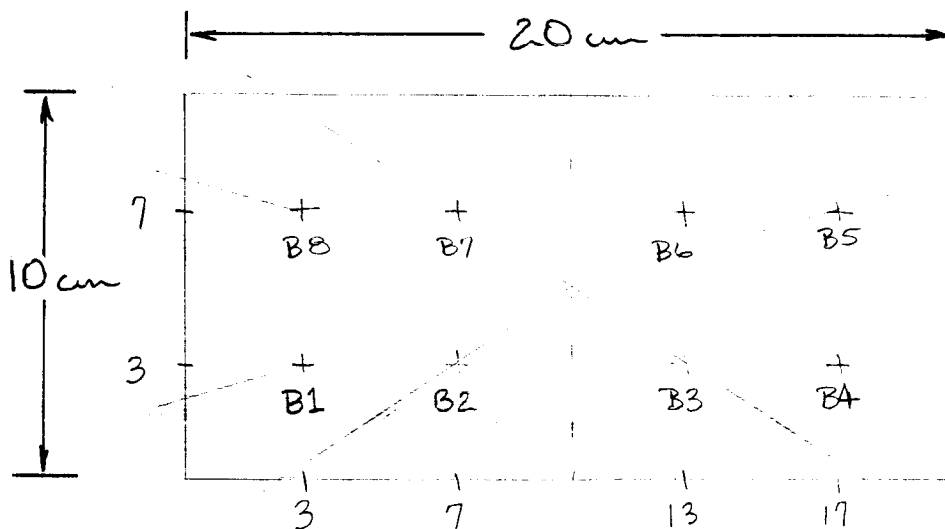
WORST CASE - MUST WITHSTAND FORCES IMPARTED BY HYDRAULIC RAM

TOTAL FORCE OF RAM = 160000 N

FORCE ON ONE SIDE = 80000 N

BOLTS AND PLATE WILL BE USED ON TOP AND BOTTOM OF I BEAM  $\Rightarrow$  CALCULATIONS WILL INSURE PROPER ADEQUATE STRENGTH.

USING BS PROGRAM ON HP-41 AND THE DIAGRAM BELOW, THE FORCES ON BOLTS WERE DETERMINED.



$F = 80000 \text{ N}$   
 $\Delta F = 2.70^\circ$   
 $x F = 1.05 \text{ m}$   
 $y F = .1 \text{ m}$

<u>BOLT</u>	<u>FORCE (N)</u>	<u>ANGLE</u>
1	199,983	107
2	95,637	127
3	112,254	-121
4	219,211	-105
5	219,211	-75
6	112,254	-59
7	95,637	53
8	199,983	73

# FRAME

DETERMINE RADIUS OF BOLT BASED ON LARGEST LOAD FORCE OF 220,000 N.

$$\tau = \frac{F}{A} \Rightarrow A = \frac{F}{\tau} \Rightarrow \pi r^2 = \frac{F}{.577 \sigma_y}$$

$$\pi r^2 = \frac{220000 \text{ N}}{.577 (4.8 \times 10^8 \text{ N/m}^2)} \Rightarrow r = 1.6 \text{ cm} \Rightarrow d = 3.2 \text{ cm}$$

DETERMINE THICKNESS OF PLATE NEEDED TO PREVENT FAILURE BY SHEAR.

$$\tau = \frac{\mu F}{A} \Rightarrow \pi r t = \frac{\mu F}{.577 \sigma_y}$$

$$\pi (1.6 \text{ cm}) t = \frac{2.5 (220000 \text{ N})}{.577 (4.8 \times 10^8 \text{ N/m}^2)}$$

$$t = 3.95 \text{ cm} \Rightarrow \text{USE } t = 4.0 \text{ cm}$$

DETERMINE THICKNESS OF PLATE NEEDED TO PREVENT SHEAR ACROSS PLANES SHOWN IN BOLT ARRANGEMENT DIAGRAM. PLANES ARE PERPENDICULAR TO ANGLE OF RESULTANT FORCE ON EACH BOLT AS DETERMINED BY AP-41 BS PROGRAM. THE DISTANCE OF TWO BOLT HOLES TAKEN FROM EACH PLANE BEFORE CALCULATION OF THICKNESS.

$$\text{BOLTS } 1 \text{ \& } 8 \quad A = \frac{\mu F}{.577 \sigma_y} \Rightarrow dt = \frac{\mu F}{.577 \sigma_y}$$

$$d = \frac{20}{\cos 17} - 6.36 = 14.55 \text{ cm}$$

$$14.55 t = \frac{2.5 (220000 \text{ N})}{.577 (4.8 \times 10^8 \text{ N/m}^2)} \Rightarrow t = \frac{1.24}{1.36} \text{ cm}$$

BOLTS 2 \& 7

$$d = \frac{10}{\sin 37} - 6.36 = 10.26 \text{ cm}$$

$$10.26 t = \frac{2.5 (220000 \text{ N})}{.577 (4.8 \times 10^8 \text{ N/m}^2)} \Rightarrow t = .84 \text{ cm}$$

# FRAME

BOLTS 3 $\frac{1}{2}$

$$\frac{10}{\sin 31} - 6.36 = 13.06 \text{ cm}$$

$$(13.06)t = \frac{2.5(112000 \text{ N})}{.577(4.8 \times 10^8 \text{ N/m}^2)} \Rightarrow t = .77 \text{ cm}$$

BOLTS 4 $\frac{1}{2}$

$$\frac{20}{\cos 15} - 6.36 = 14.35 \text{ cm}$$

$$14.35t = \frac{2.5(220000 \text{ N})}{.577(4.8 \times 10^8 \text{ N/m}^2)} \Rightarrow t = 1.38 \text{ cm}$$

SHEAR IS LIMITING

$$\Rightarrow t = 4.0 \text{ cm}$$

SELECTION OF BOLTS.

MATERIAL	AL 2024-T86
DIAMETER	1 $\frac{1}{4}$ "
LENGTH	5"
THREADS	8

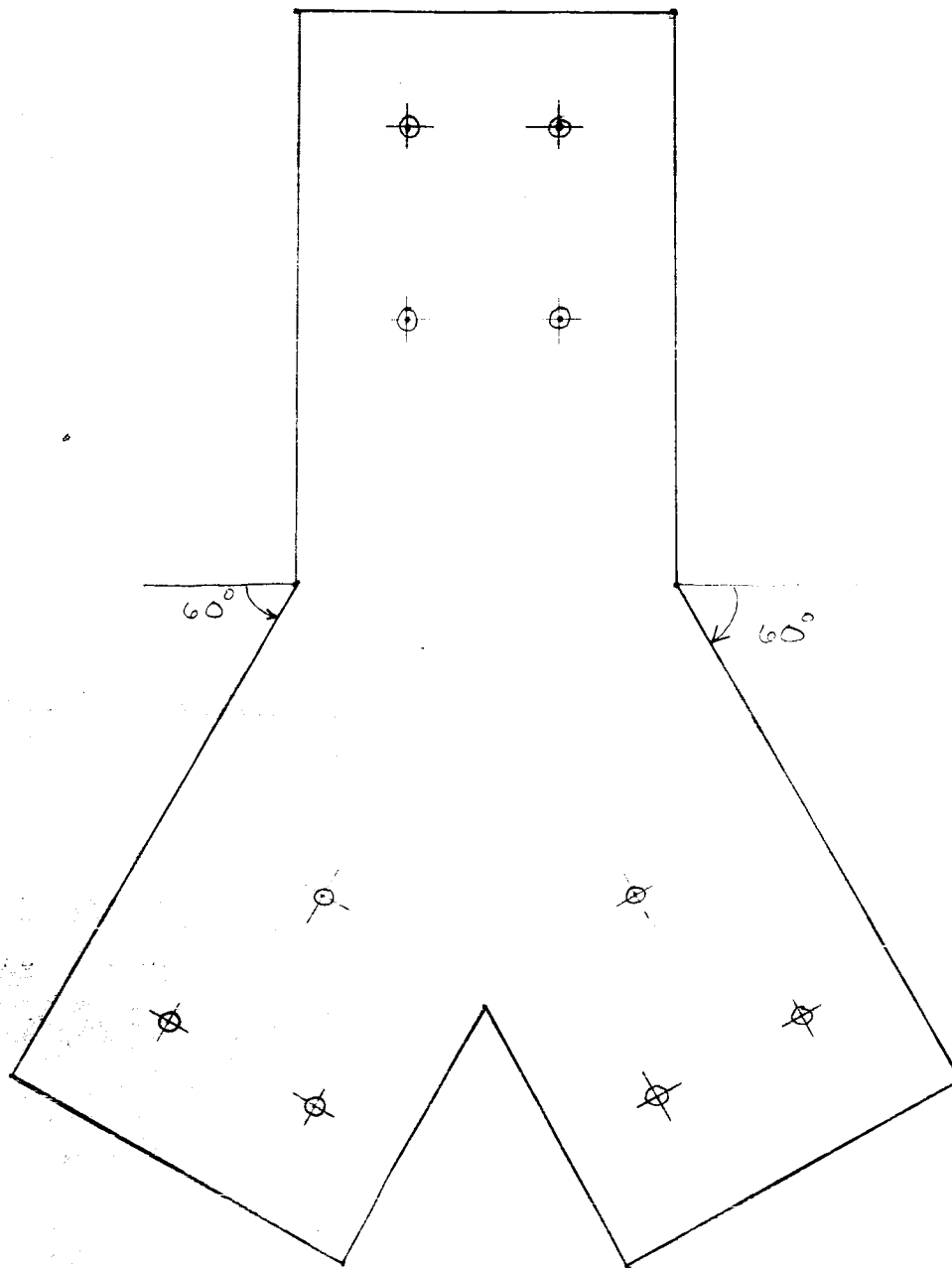
# FRAME

## FORWARD BRACE BRACKET

⊕ - BOLT LOCATIONS

SCALE:

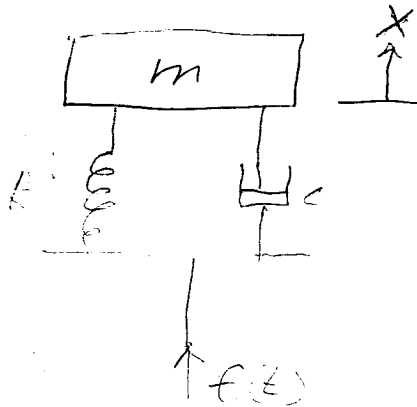
1 Sq = 1 cm



# SUSPENSION

Finding spring const. and damping const.

The system:



$$m_{max} = 26,000 \text{ N} \left( \frac{6}{9.8} \right)$$

$$m_{max} = 14,080 \text{ Kg}$$

is given by

$f(t)$  is a square wave, -40k N to 20k N, or 0 to 52,000 N

Want in compression to be 20 cm

Applied Forces function:

$$f(t) = 20,000 \sin t + 20,000$$

$$k = \frac{F_{max}}{x_{max}} = \frac{52,000 \text{ N}}{0.2 \text{ m}} = 260 \frac{\text{KN}}{\text{m}}$$

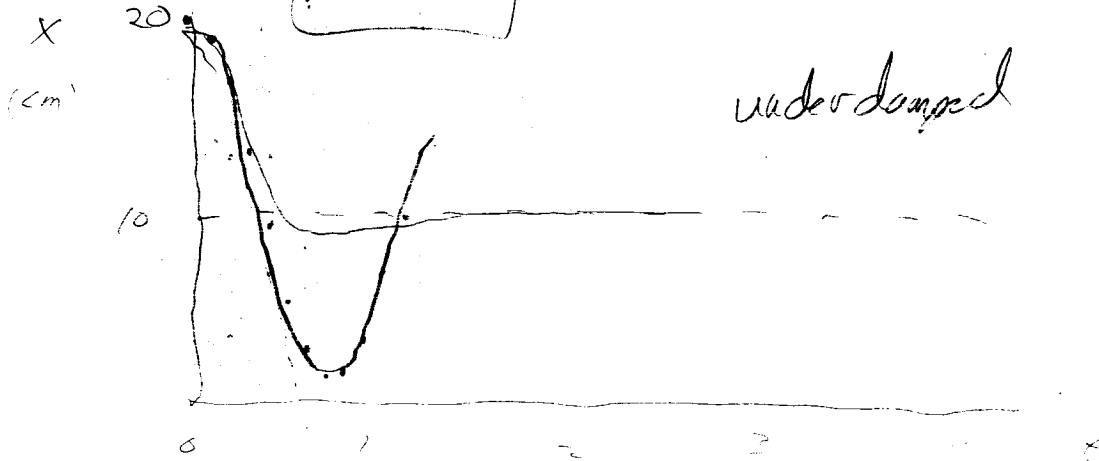
285,000

# Response to Step Input (From HP)

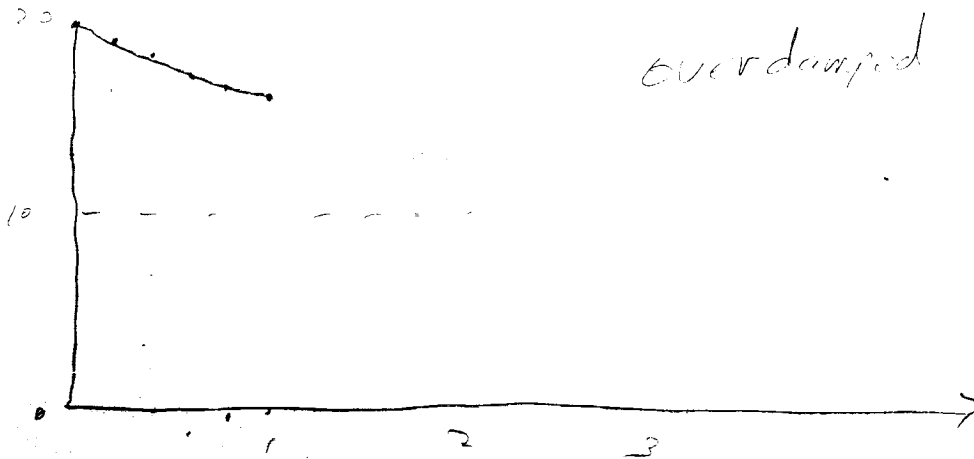
$$\zeta = 50000$$

$$K = 260,000$$

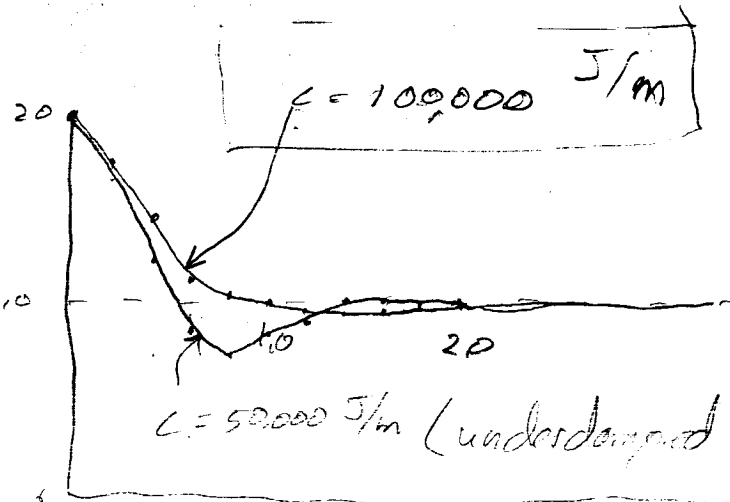
$$135 \frac{2}{4}$$



$$\zeta = 500,000 \text{ J/m}$$



$$\zeta = 100,000 \text{ J/m} \quad (\text{critically damped})$$

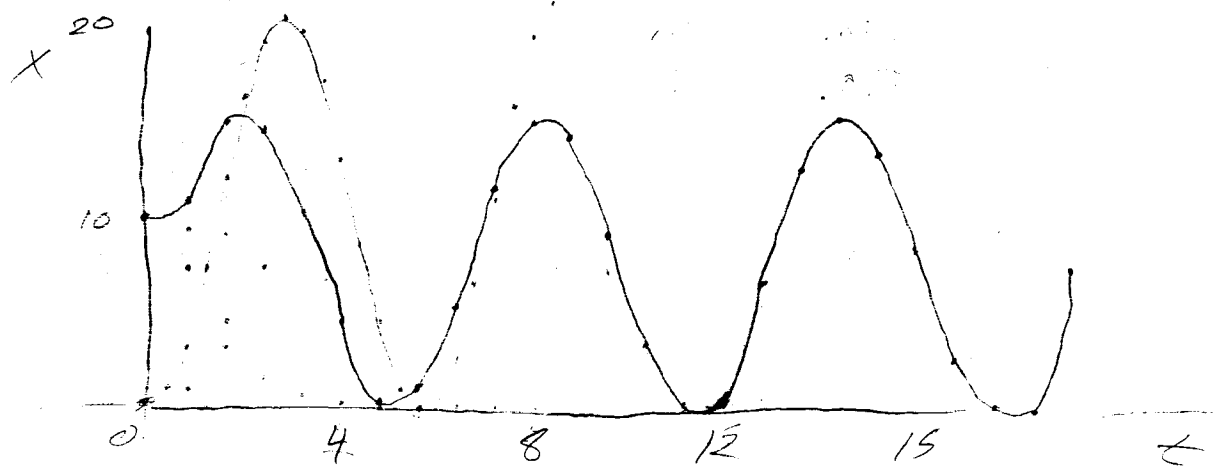


$$\zeta = 50,000 \text{ J/m} \quad (\text{underdamped})$$



Response to Harmonic Input

$$f(t) = 20,000 \sin t + 29,000 \quad \text{From HP program}$$



As shown by the graph, the response is  
 a smooth curve (bump) before  
 balancing.

$$m = 14080 \text{ kg}$$

$$F = 26,000 \text{ N} = 26,000 \text{ N}$$

$$K = 26,000 \frac{\text{N}}{\text{m}}$$

$$\frac{\text{N}}{\text{m}} = \frac{\text{L}^2}{\text{m}^2} = \frac{\text{N}}{\text{m}^2}$$

$$L = \sqrt{\frac{K \cdot m}{S}}$$

$$L^2 = N$$

$$\frac{1,000,000 \text{ In-lbs cycle}}{\text{cycle } 10 \text{ in}}$$

$$\frac{100,000 \text{ In-lbs}}{\text{in}^2}$$

$$\frac{100,000 \text{ in-lbs}}{\text{in}} = 445,000 \frac{\text{J}}{\text{min}} \cdot \frac{\text{J}}{\text{min}}$$

Spring Design HP Program  
Material: Alloy Steel ASTM-A232

From Damper:  $d_{in} \approx 4.5 \text{ in} \approx 12 \text{ cm} = 120 \text{ mm}$

$$G = 76.93 \times 10^4 \text{ N/mm}^2$$

$$@ f = 26,000; \Delta l = 20 \text{ cm}$$

$$@ f = 13,000; \Delta l = 10 \text{ cm}$$

and also:

$$\begin{aligned} d &= 1 \text{ mm} \\ L &= 10 \text{ m} \end{aligned}$$

$$50,000 \text{ J/m} = 11,240 \frac{\text{in-lbs}}{\text{in}}$$

select 2x10 bore x stroke

3 1/2 in od  
1.5 in id

4 1/4 in cyl  
1 3/8 in rod

2 1/4 in od  
1.5 in id

hole-to-hole = 2 1/2 in.

Hydraulic system

E-09-05

Gas/Air system

E-09-07

under Accumulators

E-09-15

E-09-03

E-09-05

E-09-03

E-09-05

METAL BELLOWS COPP

8577-4215

E-09-07

FORD MOTOR CO,

8502-2450

BARRY WRIGHT CO

8502-0222

E-09-15

MIDLAND BRAKE CO.

8756-0640

J-19-15

GM

8964-0008

MONROE AUTO

8962-2072

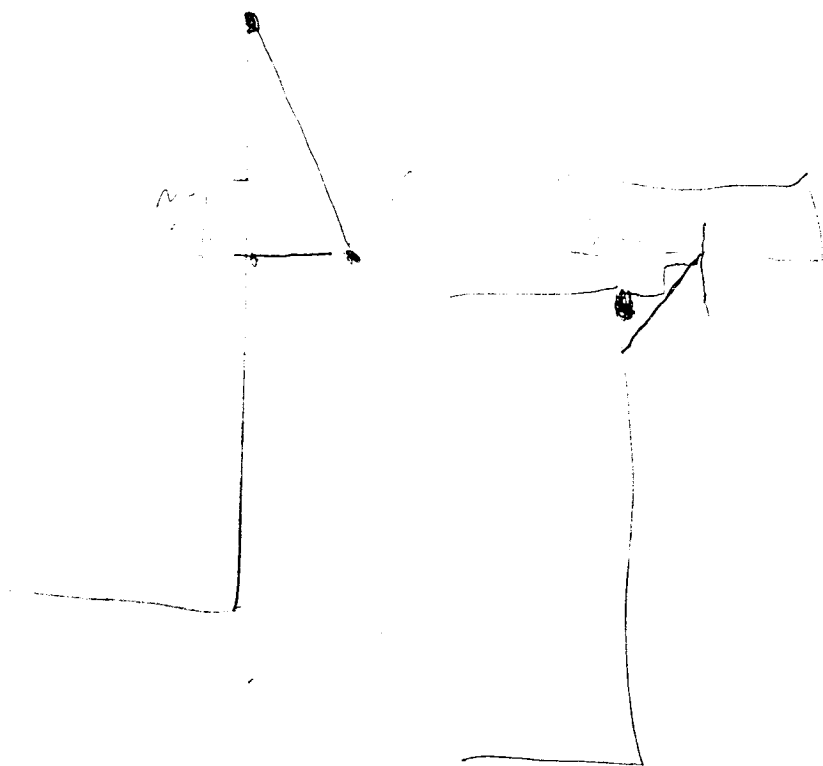
MUOG

8962-2276

CHEM PLATE

8950-2826

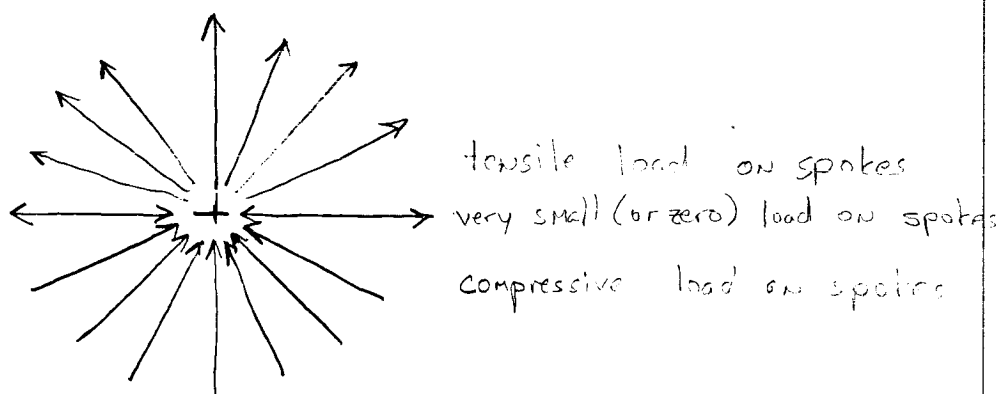
—



# Wheel Design

## Spoke Buckling

The first part of the design of the spokes is to determine the maximum axial load that a single spoke will experience. For a load of 3000 lbf (on the moon) per wheel and 20 spokes, the static diagram looks like this:



Calculating the vertical component of each spoke force and adding them, the following relations are found.

For load lined up on spokes

$$F_{load} = 12.59 F_s$$

$F_{load}$  = design load

For load lined up between spokes

$F_s$  = individual spoke load

$$F_{load} = 12.78 F_s$$

From these equations, the maximum expected load per spoke (static)  $\approx$  240 lbf. To calculate the minimum dimensions to withstand buckling under load, the formula for a Euler column is used. (This formula is used for a high  $L/k$  ratio)

$$P_{cr} = \frac{C \pi^2 EI}{l^2}$$

for fixed ends on both ends of a length,  $C = 1.2$

$$I = \frac{\pi d^4}{64} \quad \text{and for Aluminum, } E = 10.3 \times 10^3 \text{ psi}$$

Using this information, the following table was compiled.

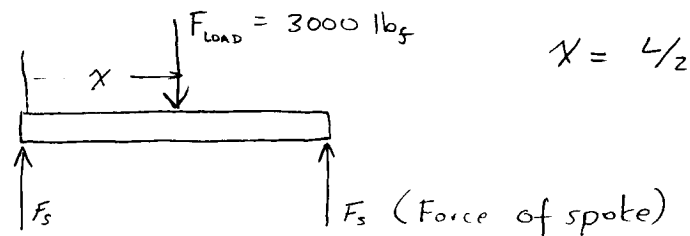
<u>Diameter (in)</u>	<u>length (in)</u>	<u>P<sub>cr</sub> (lbs)</u>
1/2"	12	2.59
1.0	12	41.58
2.0	12	665.3
3.0	12	3362.4
4.0	12	10645.3
1/2	24	0.6497
1.0	24	10.396
2.0	24	166.33
3.0	24	842.06
4.0	24	2661.33
1.0	18	18.48
2.0	18	295.70
3.0	18	1497.0
4.0	18	4731.25
2.5 (6.35cm)	18 (45.72cm)	721.9
2.36 (6cm)	12.99 (33cm)	1100.8

The last set is the dimensions that were used for the spokes. For the given load, the safety factor for

buckling is 4.6. The higher safety factor is to take into account the impact loading that will occur when the vehicle is in motion

### RIM DESIGN

The model used to represent the worst case loading at the rim is that of a beam supported at both ends with a concentrated load at the middle of the beam. The diagram is below.



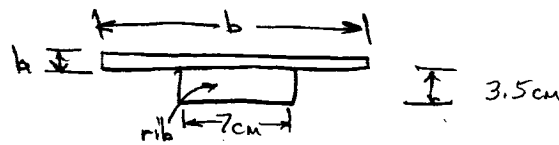
This model is justified because of the small amount of arc that is present between two adjacent spokes. The bending moment for this situation at the midpoint is equal to 5565 lbf·in. Using the equation for stress in a beam,

$$\sigma = \frac{Mc}{I}, \quad I = \frac{bh^3}{12}, \quad c = \frac{h}{2}$$

the following chart is constructed.

<u>base (in)</u>	<u>height (in) (thickness)</u>	<u>maximum stress</u>
6	1/2	44,520.0
8	1/4	133,560
8	1/2	33,390.0
10	1/2	26,712.0

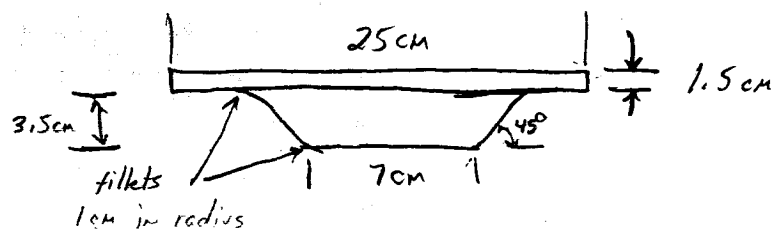
Since the stresses are so high, so the addition of an internal circumferential rib on the inside surface. The cross-sectional shape and rib dimensions are shown below.



Using this basis,  $I' = \frac{bh^3}{12} + bH(d^2)$  and  $C = \frac{b}{2} + \frac{3.5}{2}$  a new chart is constructed.

<u>b (in)</u>	<u>h (in)</u>	<u>Maximum stress</u>
6	$\frac{1}{2}$	220.97
8	$\frac{1}{4}$	111.4
8	$\frac{1}{2}$	220.25
10	$\frac{1}{2}$	219.5

The final dimensions were chosen based primarily on the need for the wider footprint than for failure protection, since the stresses experienced by the rim are about .01 the yield stress of AL-95056-O. The ribs actual crosssection was smoothed out to reduce the stress concentrations at the square corners. The final cross section is below.



## Wheel Design

### Hub Design

The minimum hub radius required is determined by the desired lateral distance between spokes. With 6cm diameter spokes, the space between the spoke bases had to be large enough to allow forging die components to be able to get between the spokes so that they could be fully formed. The dimension chosen was 32cm radius, which allows for approximately a 4cm clearance between spokes. This is enough to allow tool pieces to get between the spokes without excessive tool wear due to high stresses caused by the surface of the tool being too small.

The design of the hub also included a hole for the axle and a space for the bearings.

# Wheel Design

## Fatigue Analysis - Spokes

Spokes are loaded in compressive loading at the bottom and is in tension at the top half of the load, with the limits going from a maximum of approximately 240 lbf tension to 240 lbf compressive. The loading resembles a sinusoidal curve of 240 lbf amplitude with a frequency equal to the rpm of the wheel. This loading is similar to that of a laterally loaded rotating beam, and thus the fatigue equations for this will be used

$$S_e = K_a K_b K_c K_d K_e K_f S_e'$$

from Shigley

$$K_a = 0.78 \quad (\text{see below})$$

$$K_b = 1.0 \quad (\text{see below})$$

$$K_c = 0.814 \quad (\text{for } 99\% \text{ reliability})$$

$$K_d = 1.0 \quad (\text{assumed operating temp less than } 450^\circ\text{C})$$

$$K_e = 1.0 \quad (\text{notch-free sample})$$

$$K_f = 1.0 \quad (\text{no miscellaneous effects})$$

$$\text{for axial loading} \Rightarrow S_e' = 19.2 + 0.314 S_{uc} \quad S_{uc} \geq 60 \text{ (Kpsi)}$$

Aluminum A95056-O is the alloy that will be used in creating the wheels. It is chosen because of its high yield, tensile, and fatigue strengths combined with excellent ductility, which will be required for the wheels to be forged. (the cold-working involved in forging the wheels will decrease the ductility and increase the strengths further).

For ALuminum A 95056-0 alloy, the following values apply

Yield strength  $S_y$  22 kpsi

Tensile strength  $S_u$  42 kpsi

Shear modulus of rupture  $S_{su}$  26 kpsi

Fatigue strength  $S_f$  20 kpsi for  $50(10^7)$  cycles

Elongation in 2in, % 35

The determination of  $K_a$  is as follows:

forging results in a roughness of  $125-32 \mu\text{in}$ , usually from pg 291 Shigley, Fundamentals of Mechanical Design, the resulting value for  $K_a \approx 0.78$  at lowest value

$$S_e = (0.78)(1.0)(0.8111)(1.0)^3 S_e'$$

$$\text{Using } S_f = S_e' = 20 \times 10^3 \text{ psi}$$

$$S_e = (0.6348) 20 \times 10^3 \text{ psi}$$

$$= 12.7 \text{ Kpsi fatigue strength for } 50(10^7) \text{ cycles}$$

For the 3000 lbf load the wheels were designed for, (the area of the spoke is  $4.38 \text{ in}^2$ ) the load is about 54.7 psi. this gives a fatigue safety factor of 232.2. While this may seem to be a case of overdesigning, there are many approximations involved in the calculations. Also, many of the experiments were performed using steel instead of aluminum. Aluminum reacts differently to fatigue than steel does in that there is no lower limit of fatigue strength with increasing cycles. Thus, because the fatigue strength continues to decrease, the large initial safety factor is acceptable.

Since  $50(10^7)$  cycles corresponds to  $25(10^7)$  rotations (one rotation corresponds to 2 cycles of loading from compression to tension) the total distance traveled corresponding to  $50(10^7)$  cycles can be calculated. The outside radius of the wheel is 439.823 cm (173.16 in).

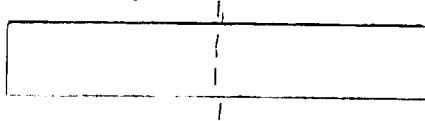
Total distance traveled is  $(439.823 \text{ cm})(25 \times 10^7) = 1.0996 \times 10^8 \text{ cm}$  which corresponds to  $1.10 \times 10^7 \text{ km}$ . A recommendation that will be made is that the wheels be pulled and inspected for fatigue damage (and other damage due to prolonged exposure in a radiation environment) every 750,000 km of travel or every 5 years of use, whichever comes first.

# Wheel Design

## Construction

The wheel will consist of a one piece Aluminum forging made of AL-95056-O alloy. Forging was chosen for the strength of the wheel after manufacture and for the ease of construction without excessive machining. The probable steps that would be involved in the creation of the wheel are as follows:

1. Starting Blank

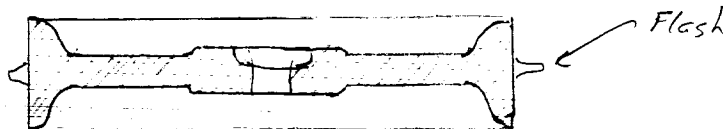


2. First Shape Forging (gives general shape)

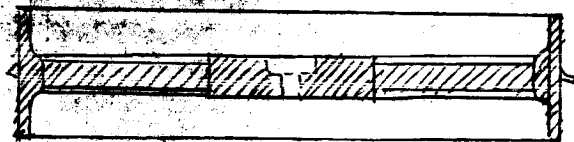


3. Second Shape Forging

Forms spokes and axle clearance



4. Final Forging gives completed shape



5. Final Machining  
remove flash, polish hub riding surfaces

More intensive research and development may discover exactly what processes will be involved.

Several Programs were written on the HP-41CV to help process and tabulate the tables earlier in this appendix. They are as follows.

#### Program "ME"

```

01 LBL TME
02 LBL 01
03 T H = ?
04 PROMPT
05 STO 01
06 T B = ?
07 PROMPT
08 STO 02
09 RCL 01
10 3
11 Y  $\uparrow$  X
12 RCL 02
13 *
14 12
15 /
16 6,2334
17 +
18 STO 03
19 RCL 01
20 2
21 /
22 STO 04
23 RCL 04
24 5565.0
25 *
26 RCL 03
27 /
28 T STRESS =
29 RCL X
30 AVEIW
31 STOP
32 GTD 01
33 END

```

Note: program used to determine stress in rim section with rib

#### PROGRAM "MEZ"

```

01 LBL TMEZ
02 LBL 01
03 T D = ?
04 PROMPT
05 4
06 Y  $\uparrow$  X
07 5988
08 *
09 T L = ?
10 PROMPT
11 X  $\uparrow$  Z
12 /
13 T PCR =
14 ARCL X
15 AVEIW
16 STOP
17 GTD 01
18 END

```

Note: program used to determine critical load of buckling for spokes

## Hydraulics Calculations

Because standard dimensions for anything dealing with hydraulics is in English units, all calculations will be done in English units.

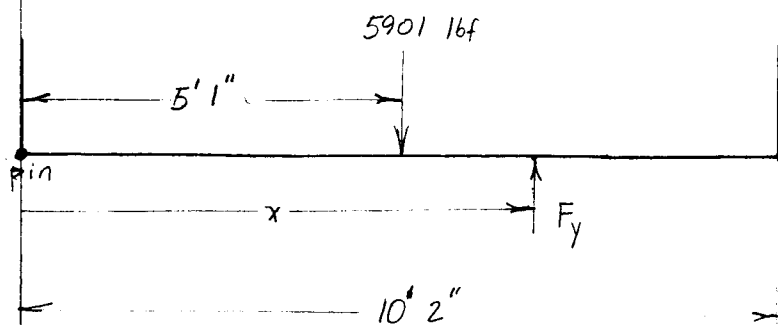
$$\text{weight of bed} = 1700 \text{ N}$$

$$\text{weight of load} = 24500 \text{ N}$$

$$\text{total weight} = \underline{26200 \text{ N}}$$

$$\text{weight: } (26200 \text{ N}) \left( \frac{1 \text{ lbf}}{4.44 \text{ N}} \right) = 5901 \text{ lbf}$$

Determination of forces required to lift bucket



assuming a load of 5901 lb<sub>f</sub> must be lifted the amount of force required to lift (F<sub>y</sub>) can be found at varying distances of x.

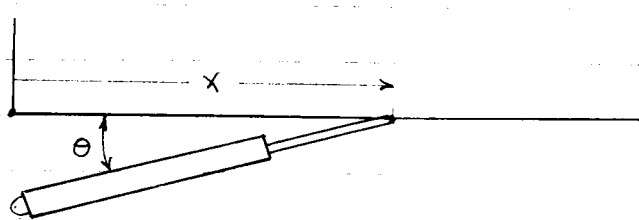
$$\sum M_{pin} = 0 = -30000 \text{ lb}_f \cdot \text{ft} + F_y(x) = 0$$

<u>x (ft)</u>	<u>F<sub>y</sub> (lb<sub>f</sub>)</u>
1	30,000.0
2	15,000.0
3	10,000.0
4	7,500.0
5	6,000.0
6	5,000.0
7	4,286.0
8	3,750.0
9	3,333.3
10	3,000.0

choose x of 4, 5, 6, 7, 8 as optimum lengths

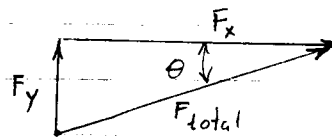
# Determination of Hydraulic ram placement

if the following setup is used to raise the bucket calculations can be made for  $F_{total}$  required to lift bucket



a free body diagram

$$F_{total} = \frac{F_y}{\sin \theta}$$



$F_{total}$  Calculation

$\theta \backslash x$	4'	5'	6'	7'	8'
5°	86053	68842	57369	49176	43026
10°	43191	34553	28794	24682	21595
15°	28978	23182	19319	16560	14489
20°	21929	17543	14619	12531	10964

choose 6' as place for ram to connect to bed at angle of 10°

$\therefore$  cylinder that produces 28794 lb of force is required.

## Calculation of hydraulic size required.

In choosing a hydraulic ram general practice calls for a ram which will produce 20%-25% more force than is required. This will give a safety factor to take care of pressure losses in valves, piping, flow control valves, and mechanical friction in the cylinder.

A ram capable of  $(1.25)(28794) = 35993$  lbf is required.

$$\text{Force} = (\text{pressure})(\text{area})$$

$$\text{area} = \pi r^2$$

$$\pi r^2 = \frac{35993 \text{ lbf}}{3000 \text{ psi}} = 11.9977$$

$$r = 1.9542 \quad \text{diameter} = 3.9084$$

choose a 4" bore for the hydraulic ram

from catalog a 4" bore matches with a rod diameter of 2.5"

Rod Buckling

$$l = 2' = 24 \text{ in}$$

$$k = \frac{d}{4}$$

$$\left(\frac{l}{k}\right)_1 = \sqrt{\frac{2\pi^2 CE}{S_y}} = \sqrt{\frac{(2)(\pi)^2(2)(30 \times 10^6)}{100,000}} = 108.83$$

$$\frac{l}{k} = \frac{24}{\frac{4}{4}} = 24$$

since  $\frac{l}{k} < \left(\frac{l}{k}\right)_1$  use Johnson

$$\frac{P_{cr}}{A} = S_y - b \left(\frac{l}{k}\right)^2$$

$$b = \left(\frac{S_y}{2\pi}\right)^2 \frac{1}{CE}$$

$$b = \left(\frac{100,000}{2\pi}\right)^2 \left(\frac{1}{(2)(30 \times 10^6)}\right) = 4.222$$

$$P_{cr} = (\pi)(1.25)^2 (100,000 - 4.222(24)^2) = 478,936.4 \text{ psi}$$

$$\text{actual } P \text{ at extension} = \frac{35993 \text{ lb}_f}{\pi(1.25)^2} = 7332.4 \text{ psi}$$

since  $P < P_{cr}$  rod diameter is acceptable.

## Cylinder velocity

$$\text{velocity (ft/sec)} = \frac{231 \times \text{flow rate (gpm)}}{12 \times 60 \times \text{area (in}^2\text{)}}$$

$$= \frac{231 \times 20}{12 \times 60 \times (\pi)(2)^2}$$

$$= .5106 \text{ ft/sec}$$

if a stroke of 24" is required the time to extend is as follows:

$$(2 \text{ ft}) \frac{1 \text{ sec}}{.5106 \text{ ft}} = 3.92 \text{ sec}$$

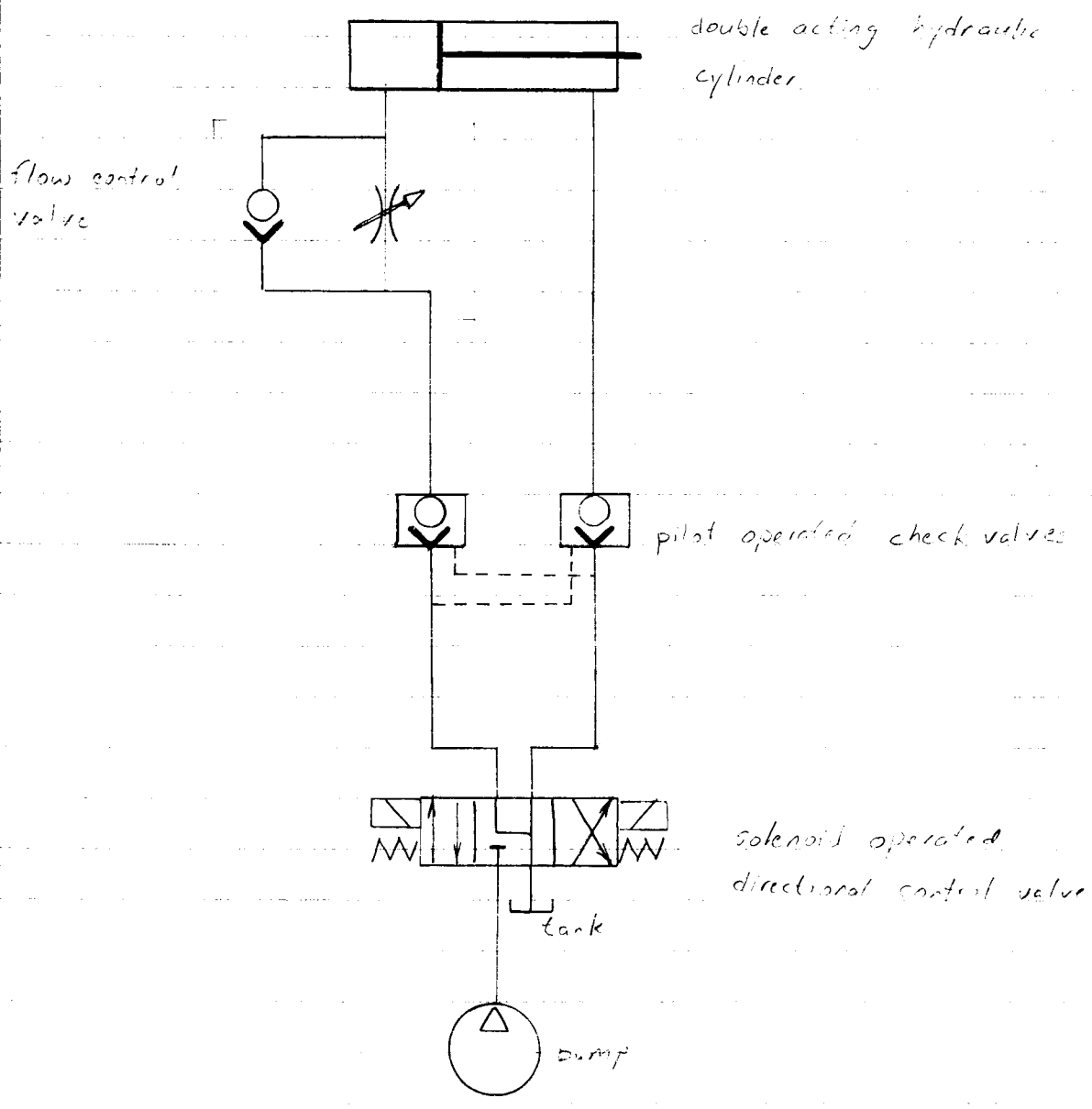
$$\boxed{\text{time to extend} = 3.92 \text{ sec}}$$

$$V = \frac{231 \times 20}{12 \times 60 \times (\pi)(2^2 - 1.25^2)} = .8379$$

$$\boxed{\text{time to retract} = 2.39 \text{ sec}}$$

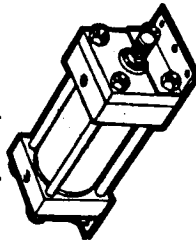
a flow control valve will be used to increase retraction time

# Hydraulic circuit

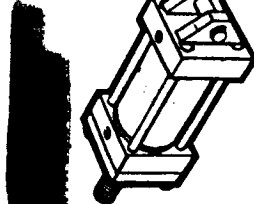
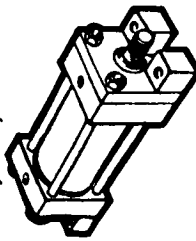


Side End Angles, Side End Lugs  
and Cap Fixed Clevis Mountings  
1 1/2" to 8" bore sizes

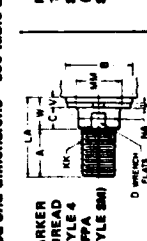
Side End Angles Mounting  
Parker Style CB  
(NFPA Style MS1)



Side End Lugs Mounting  
Parker Style G  
(NFPA Style MS7)



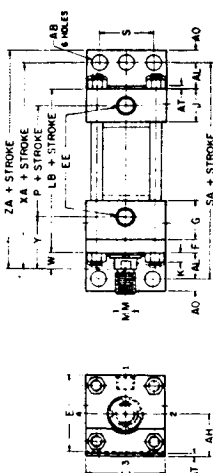
Side End Angles Mounting  
Parker Style CB  
(NFPA Style MS1)



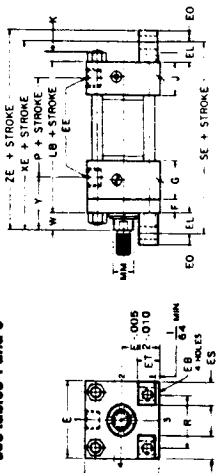
A rod and high strength stud is supplied on threaded styles 4 and 5. The rod end is recommended where the workpiece is secured against the rod shoulder. When the workpiece is not supported, style 4 will be supplied.

Parker Series 2H  
Heavy Duty Hydraulic Cylinders  
1 1/2" to 8" bore sizes

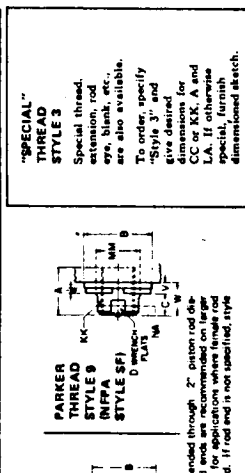
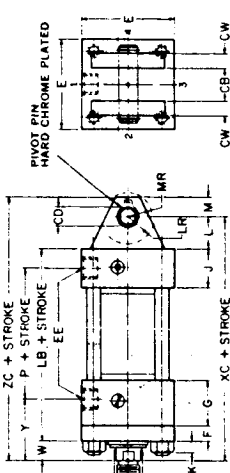
Envelope and mounting dimensions—  
see tables 1 and 3



Envelope and mounting dimensions—  
see tables 1 and 3



Envelope and mounting dimensions—  
see tables 1 and 3



**"SPECIAL" THREAD STYLE 3**  
Special thread, extension, rod end, and mounting dimensions are also available. To order, specify "Special" thread, extension, rod end, and mounting dimensions desired. CC or KK, A and L.A. If otherwise specified, furnish dimensioned sketch.

Parker Series 2H  
Heavy Duty Hydraulic Cylinders  
1 1/2" to 8" bore sizes

Table 1—Envelope and mounting dimensions

Bore	AS	AM	AL	AO	AT	CC	EW	IE	IB	IS	IT	J	K	L	LA	MA	N	S	LE	P	SA	SE
1 1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
2 1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
3	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
3 1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
4	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
5	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
6	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
7	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2
8	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2

\* SAE straight thread bolts are standard and are indicated by part number.  
† Head end cushions are non-adjustable in 1/4".  
‡ 2" and 2 1/2" bore cylinders with 1/2" rod.

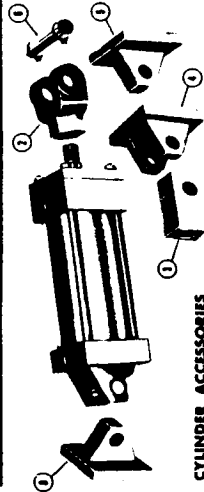
Table 2—Rod dimensions

ROD EXTENSIONS AND PILOT DIMENSIONS										ADD STROKE																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																				
Bore	Rod No.	Rod Dia.	Thread	ROD EXTENSIONS AND PILOT DIMENSIONS										ADD STROKE																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																
				CC Inch	EW Inch	A Inch	B Inch	C Inch	D Inch	LA Inch	NA Inch	V Inch	W Inch	Y	ZA	XC	XB	ZE																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																												
1 1/2	1 (Std.)	1/2	2 1/2-12	3/4	2 1/2-12	3/4	1 1/4	3/4	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	1/2	

CAUTION: When using mounting styles CB and G, check clearance between mounting members and rod attachment or necessary. If necessary, specify longer rod extension to avoid interference with mounting members.  
NOTE: Shaded areas indicate non-standard rod sizes that are available and made to order.

## Cylinder Accessories

## Parker Series 2H Heavy Duty Hydraulic Cylinders



**CYLINDER ACCESSORIES**  
Parker offers a complete range of cylinder accessories to assure you of greatest versatility in present or future cylinder applications.

**ROD END ACCESSORIES**  
Accessories offered for the rod end of the cylinder include Rod Clevis, Eye Bracket, Knuckle, Clevis Bracket and Pivot Pin. To select the proper part number for any desired accessory, refer to Chart A below and look opposite the thread size of the rod end as indicated in the first column. The Pivot Pins, Eye Brackets and Clevis Brackets are listed opposite the thread size which their mating Knuckles or Clevises fit.

CHART A		CHART B	
THREAD SIZE	ROD END ACCESSORY	THREAD SIZE	ROD END ACCESSORY
1/4"-20	50940	1/4"-20	50940
3/8"-16	50941	3/8"-16	50941
1/2"-13	50942	1/2"-13	50942
3/4"-10	50943	3/4"-10	50943
1"-8	50944	1"-8	50944
1 1/4"-6	50945	1 1/4"-6	50945
1 1/2"-5	50946	1 1/2"-5	50946
2"-4	50947	2"-4	50947
2 1/2"-3 1/2	50948	2 1/2"-3 1/2	50948
3"-3	50949	3"-3	50949
3 1/2"-2 1/2	50950	3 1/2"-2 1/2	50950
4"-2	50951	4"-2	50951
4 1/2"-1 1/2	50952	4 1/2"-1 1/2	50952
5"-1	50953	5"-1	50953
5 1/2"-3/4	50954	5 1/2"-3/4	50954
6"-3/8	50955	6"-3/8	50955
6 1/2"-1/2	50956	6 1/2"-1/2	50956
8"-3/8	50957	8"-3/8	50957
10"-3/8	50958	10"-3/8	50958
12"-3/8	50959	12"-3/8	50959
14"-3/8	50960	14"-3/8	50960
16"-3/8	50961	16"-3/8	50961
18"-3/8	50962	18"-3/8	50962
20"-3/8	50963	20"-3/8	50963
24"-3/8	50964	24"-3/8	50964
30"-3/8	50965	30"-3/8	50965
36"-3/8	50966	36"-3/8	50966
42"-3/8	50967	42"-3/8	50967
48"-3/8	50968	48"-3/8	50968
54"-3/8	50969	54"-3/8	50969
60"-3/8	50970	60"-3/8	50970
72"-3/8	50971	72"-3/8	50971
84"-3/8	50972	84"-3/8	50972
96"-3/8	50973	96"-3/8	50973
108"-3/8	50974	108"-3/8	50974
120"-3/8	50975	120"-3/8	50975
144"-3/8	50976	144"-3/8	50976
168"-3/8	50977	168"-3/8	50977
192"-3/8	50978	192"-3/8	50978
216"-3/8	50979	216"-3/8	50979
240"-3/8	50980	240"-3/8	50980
264"-3/8	50981	264"-3/8	50981
288"-3/8	50982	288"-3/8	50982
312"-3/8	50983	312"-3/8	50983
336"-3/8	50984	336"-3/8	50984
360"-3/8	50985	360"-3/8	50985
384"-3/8	50986	384"-3/8	50986
408"-3/8	50987	408"-3/8	50987
432"-3/8	50988	432"-3/8	50988
456"-3/8	50989	456"-3/8	50989
480"-3/8	50990	480"-3/8	50990
504"-3/8	50991	504"-3/8	50991
528"-3/8	50992	528"-3/8	50992
552"-3/8	50993	552"-3/8	50993
576"-3/8	50994	576"-3/8	50994
600"-3/8	50995	600"-3/8	50995
624"-3/8	50996	624"-3/8	50996
648"-3/8	50997	648"-3/8	50997
672"-3/8	50998	672"-3/8	50998
696"-3/8	50999	696"-3/8	50999
720"-3/8	51000	720"-3/8	51000

\*For alignment coupler dimensions, see page 140.

### ACCESSORY LOAD CAPACITY

The various accessories on Pages 64 and 65 have been load rated for your convenience. The load capacity in lbs. shown on page 65 is the recommended maximum load for that accessory based on a 4:1 design factor in tension. (Pivot pin is rated in shear.) Before specifying, compare the actual load or the tension (pull) force at maximum operating pressure of the cylinder with the load capacity of the accessory you plan to use. If load or pull force of cylinder exceeds load capacity of accessory, consult factory.

PLATE	SERIES "2H"
BB	1 1/2"
BB	2"
BB	2 1/2"
BB	3"
BB	4"
BB	5"
BB	6"
BB	8"

## Parker Series 2H Heavy Duty Hydraulic Cylinders

Female Rod Clevis Part Number		Knuckle Part Number		Clevis Bracket for Knuckle Part Number		Eye Bracket and Mounting Plate Part Number		Pivot Pin Part Number	
512111	50940	50941	50942	50943	50944	50945	50946	50947	50948
A	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CB	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CD	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CE	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CW	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
ER	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
KK	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
LR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
MR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
RR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
512112	50940	50941	50942	50943	50944	50945	50946	50947	50948
A	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CB	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CD	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CE	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CW	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
ER	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
KK	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
LR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
MR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
RR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
512113	50940	50941	50942	50943	50944	50945	50946	50947	50948
A	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CB	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CD	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CE	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
CW	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
ER	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
KK	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
LR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
MR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"
RR	1/4"	3/8"	1/2"	3/4"	1"	1 1/4"	1 1/2"	1 3/4"	2"

See Accessory Load Capacity note on page 64.  
These sizes are standard with cotter pins.  
† INCLUDES PIVOT PIN

# Fluidpower Components



## Flow Control Valves Series F and FS

### Colorflow Series F and FS Flow Control Valves

These valves provide precise control of flow and shut-off in one direction, and automatically permit full flow in the opposite direction.

A two-step needle allows fine adjustment at low flow by using the first three turns of the adjusting knob; the next three turns open the valve to full flow, and also provide standard throttling adjustments.

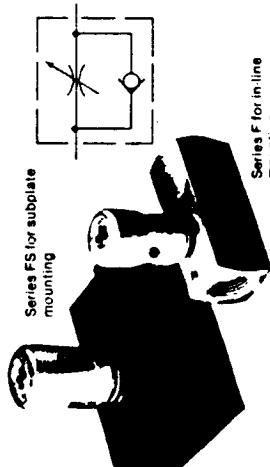
The exclusive "Colorflow" color-band reference scale on the valve stem is a great convenience and time-saver in setting the valve originally and in returning it to any previous setting.

Available in brass for air and oil applications.

Maximum operating pressure:

Brass: Models F, inline only: 2000 PSI (140 Bar), max. except Model F1600 only: 500 PSI (35 Bar) max.

Steel: Inline 200-820 Sizes: 5000 PSI  
Inline 1020-3220 Sizes: 3000 PSI  
Subplate All Sizes: 3000 PSI



Cracking Pressure, for return check poppet: 5 PSI (0.4 Bar)

Poppet material: Brass F200 through F800  
Soft seal. All others: Solid metal poppet.

Needles: Standard needle on all models; Fine needle optional on Models F400, 600, 620, 820

### Quick Reference Data Chart

Model Number	Free Flow Rate, Max. GPM (L/M)	Free Flow Orifice Area In <sup>2</sup>	Free Flow C <sub>d</sub>	Orifice Area, Effective Control Flow, In <sup>2</sup>	Effective Control Flow C <sub>v</sub>	Port Size (In.) And Threads
F200	3 (11)	0.023	0.53	0.0102	0.230	1/8 NPTF
F400	5 (19)	0.068	1.56	0.194	0.433	1/4 NPTF
F600	8 (30)	0.099	2.27	0.344	0.787	3/8 NPTF
F800	15 (57)	0.224	5.11	0.427	0.976	1/2 NPTF
F1020						1/2 NPTF
F1200						3/4 NPTF
F1220						1-1/16-12 UN (SAE 12)
F1420	25 (95)	0.348	7.95	0.500	3.420	1-3/16-12 UN (SAE 14)
F1600	40 (151)	0.453	10.35	2.300	5.250	1 NPTF
F1620	40 (151)	0.453	10.35	3.070	7.000	1-5/16-12 UN (SAE 16)
F2000	70 (265)	0.855	19.52	2.300	5.250	1-1/4 NPTF
F2020	70 (265)	0.855	19.52	3.710	8.470	1-5/8-12 UN (SAE 20)
F2400	100 (379)	0.955	21.82	2.300	5.250	1-1/2 NPTF
F2420	100 (379)	0.955	21.82	3.710	8.470	1-7/8-12 UN (SAE 24)
F3200	150 (568)	1.046	23.90	2.300	5.250	2 NPTF
F3220	150 (568)	1.046	23.90	6.010	13.410	2-1/2-12 UN (SAE 32)

\*Coast Guard Acceptance-Steel.

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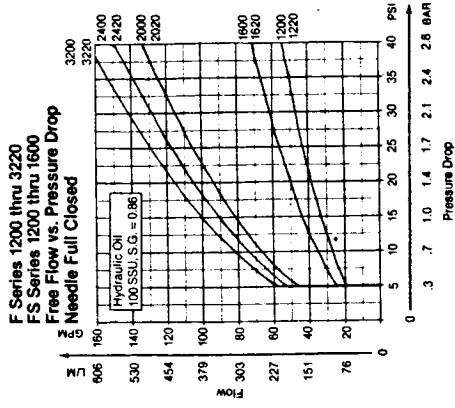
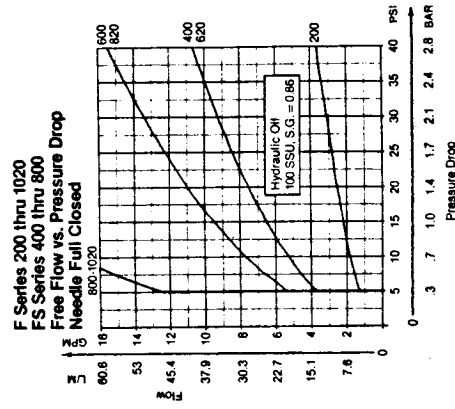
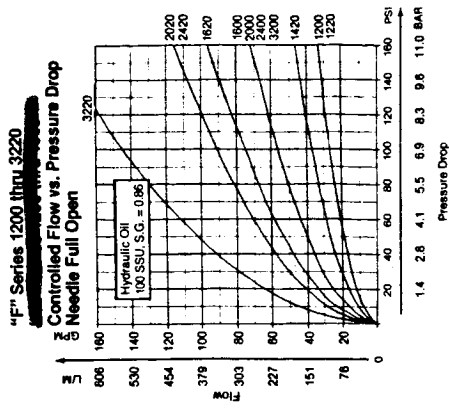
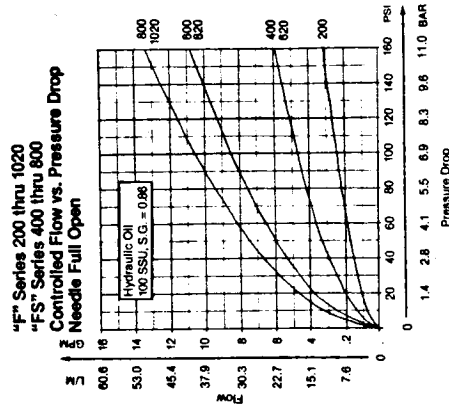
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# Fluidpower Components



## Flow Control Valves Series F and FS

### Performance



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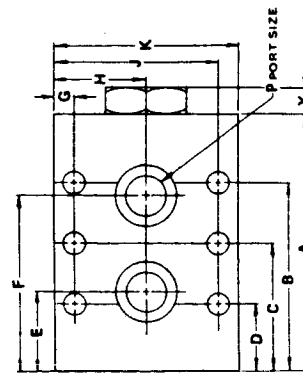
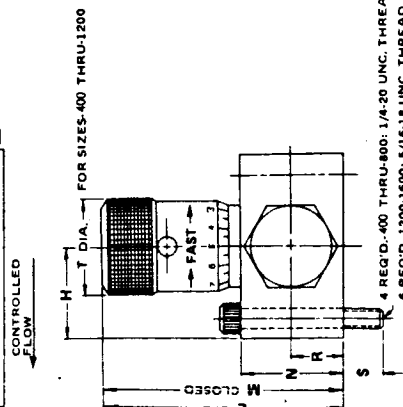
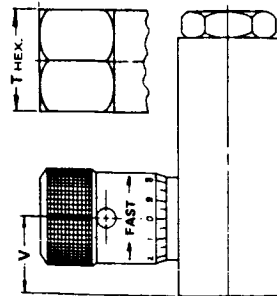
## Flow Control Valves Series F and FS

### Dimensions

#### Subplate-mounted Flow Control Valves

Models FS400 through FS1600

NOTE:  
HEX KNOB  
IS STANDARD  
ON 1600 SIZE.

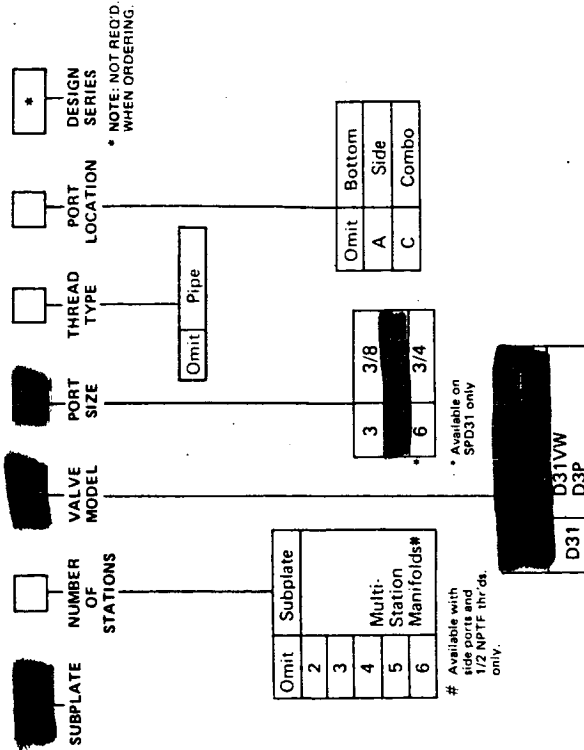


	Valve Model	
	FS400	FS600
A	2.50 (63.5)	2.75 (69.9)
B	1.94 (49.3)	2.03 (51.6)
C	—	—
D	56 (14.2)	72 (18.3)
E	75 (19.1)	88 (22.4)
F	1.75 (44.5)	1.88 (47.8)
G	22 (5.6)	25 (6.4)
H	88 (22.4)	1.00 (25.4)
J	1.53 (38.9)	1.75 (44.5)
K	1.75 (44.5)	2.00 (50.8)
L	2.21 (56.1)	2.65 (67.3)
M	2.01 (51.1)	2.40 (61.0)
N	87 (22.1)	1.00 (25.4)
P	28 (7.1)	41 (10.4)
R	43 (10.9)	50 (12.7)
S	38 (9.7)	50 (12.7)
T	81 (20.6)	1.00 (25.4)
V	84 (21.3)	1.21 (30.7)
X	31 (7.9)	32 (8.1)

## Ordering Information

### ACCESSORIES

SUBPLATES AND MANIFOLDS  
Bolt kit for mounting valves to subplate is BK226.



### AVAILABLE SUBPLATES & MANIFOLDS UNIT WEIGHTS:

SPD33	5 LBS.
SPD33A	5 LBS.
SPD33C	5 LBS.
SPD34	5 LBS.
SPD34A	5 LBS.
SPD34C	5 LBS.
SPD314	5 LBS.
SPD316	7.5 LBS.
SP2D34A	11.5 LBS.
SP3D34A	15.0 LBS.
SP4D34A	19.0 LBS.
SP5D34A	23.0 LBS.

Technical Information



ENGINEERING  
PERFORMANCE DATA

M*T.G. PATTERN NFPA D02-3/8 inch	
MAXIMUM PRESSURE	Operating 3000 PSI (205 Bar)
Tank Line	
1500 PSI (102 Bar)	
NOMINAL FLOW	12 GPM(45 LITERS/MIN.)
MAXIMUM FLOW	See chart below.

SOLENOID ELECTRICAL CHARACTERISTICS			
SOLENOID CODE	NOMINAL VOLTS/Hz	CURRENT (AMP)*	
		In Rush	Holding
Y	120/60 110/50	4.2 4.3	72 75
YF**	120/60 110/50	2.8 2.9	40 42
T	240/60 220/50	2.1 2.2	36 38
E	24/60 24/50	16 20	2.7 3.5
SOLENOID CODE	NOMINAL VOLTAGE	Current (AMP)*	
		Watts*	
L	6 VDC	6.0	36
K	12 VDC	3.0	36
D	120 VDC	.3	36
Z	250 VDC	.14	36

\*Based on nominal voltage  
\*\*3000 PSI, 12 GPM

The wet armature coil is a two lead dual frequency coil. It can be used on 50 or 60 hertz current without rewiring.

RESPONSE TIME			
Nominal Response time (milliseconds) at 3000 PSI, 12 GPM			
SOLENOID TYPE	PULL-IN	DROP-OUT	
AC	16	20	

QUICK REFERENCE DATA CHART				
Model	Spool Symbol	Maximum Flow (GPM) 3000 PSI	Model	Spool Symbol
D3W1		20	D3W5	
D3W2		20	D3W6	
D3W3		20	D3W7	
			D3W8	

Technical Information

ENGINEERING  
PERFORMANCE DATA

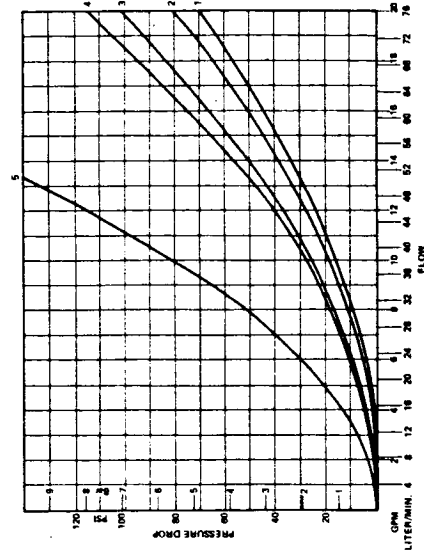
D3W SERIES PRESSURE DROP CHART

The following chart provides the flow vs. pressure drop curve reference for the D3W Series valve by spool type.

Example:  
Find the pressure drop from P to B using a D3W8 (8 spool) Series valve.

Using the D3W chart locate the numeral 8 in the Spool Code column. To the right of the numeral 8, locate the numeral 4 in the P-B column. A pressure drop (ΔP) from P to B using a D3W8 valve would be obtained on curve number 4.

D3W Pressure Drop Curve Reference Chart						
Spool Code	Curve Number					
	P-A	P-B	P-T	A-T	B-T	
1	4	4	-	3	3	
2	3	3	2	2	2	
3	4	4	-	1	3	
4	4	4	-	1	1	
5	3	4	-	3	3	
6	3	3	-	3	3	
7	4	3	5	3	2	
8	4	4	5	3	2	
10	4	4	-	-	-	
11	4	4	-	-	3	



PRESSURE DROP vs. FLOW

Curves were generated using 100 SSU hydraulic oil. For any other viscosity pressure drop will change as per chart.		VISCOSITY CORRECTION FACTOR					
Viscosity (SSU)	Percentage of Δ P (Approx.)	75	150	200	250	300	400
		93	111	119	126	132	141

## Series D3W

Solenoid Operated  
Valves, Directional Control

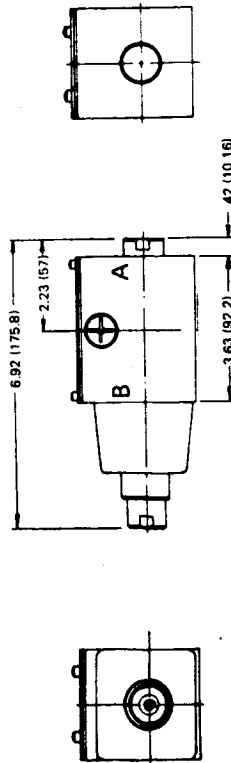
## Technical Information

### DIMENSIONS

\*MILLIMETER EQUIVALENTS FOR INCH DIMENSIONS ARE SHOWN IN (\*\*)

### AC SOLENOIDS

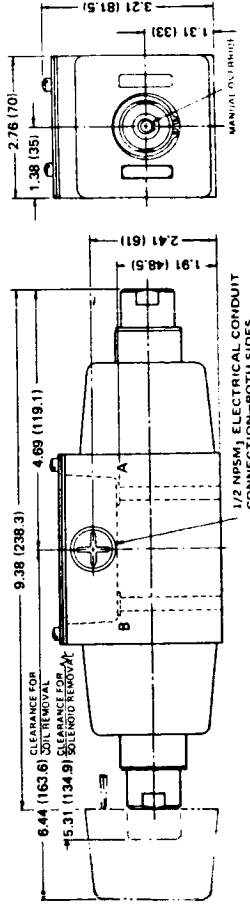
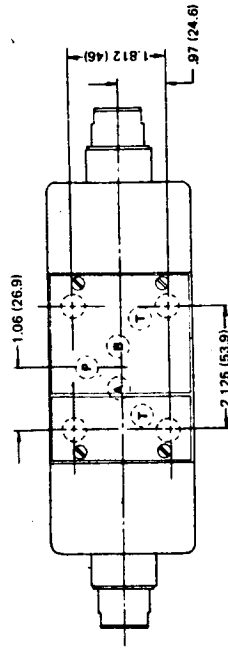
SINGLE AC SOLENOID, SPRING OFFSET MODELS  
D3W\*B\*, D3W\*E\*, D3W\*F\*, D3W\*H\*, D3W\*K\*, D3W\*M\*



NOTE: ON SINGLE SOLENOID MODELS,  
SOLENOID CAN BE MOUNTED EITHER END.

### DOUBLE AC SOLENOID, SPRING CENTERED AND DETENTED MODELS D3W\*C\*, D3W\*D\*

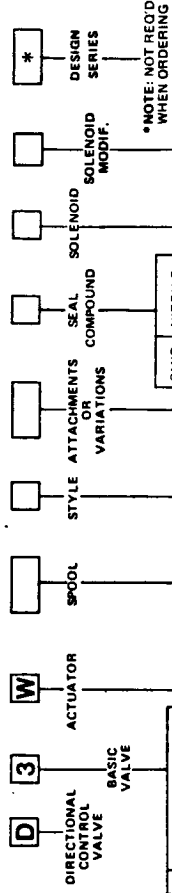
NOTE: ON VALVES WITH 8 SPOOL, A AND/OR B OPERATORS REVERSE SIDES. FLOW PATHS REMAIN THE SAME.



## Series D3W

Solenoid Operated  
Valves, Directional Control

## Ordering Information



3 3/8 NFPA D02  
SUBPLATE MOUNTING

OMIT	STANDARD VALVE - NO VARIATIONS
1	CONDUIT BOX
4	C.S.A. APPROVAL
5*	SIGNAL LIGHTS
6	MANUAL PLUG - PLUG IN
30	SPIN PLUG IN WITH SINGLE SOLENOID

\*With coils Y, T, D and Z only.

\*F LOW AMP COIL  
\*\*V solenoid only. See full data page 30-A.

\*NOTE: NOT REQ'D. WHEN ORDERING

CODE	STYLE
B*	SINGLE SOLENOID TWO POSITION, SPRING OFFSET, P = A and B = T IN SPRING POSITION
D*	DOUBLE SOLENOID TWO POSITION, DETENT
F	SINGLE SOLENOID, TWO POSITION SPRING OFFSET TO CENTER, P = B and A = T WHEN ENERGIZED
F	SINGLE SOLENOID, TWO POSITION SPRING OFFSET, ENERGIZE TO CENTER, P = A and B = T IN SPRING OFFSET POSITION
H*	SINGLE SOLENOID, TWO POSITION SPRING OFFSET, P = B and A = T IN SPRING POSITION
K	SINGLE SOLENOID, TWO POSITION SPRING OFFSET TO CENTER, P = A and B = T WHEN ENERGIZED
M	SINGLE SOLENOID, TWO POSITION SPRING OFFSET, ENERGIZE TO CENTER POSITION, P = B and A = T IN SPRING OFFSET POSITION

YY	120V/60HZ 110V/50HZ
T	240V/60HZ 220V/50HZ
E	24V/60HZ 24V/50HZ
L	8 VDC
K	12 VDC
KK	HIRSCHMANN
JJ	HIRSCHMANN
D	120 VDC
Z	250 VDC

UNIT WEIGHT:  
Single Solenoid 8.5 lbs. (3.9 kg.)  
Double Solenoid 10 lbs. (4.5 kg.)

\*Available with 1, 2, 4, 11 speeds only.  
This condition varies with spool code

## Technical Information

### ENGINEERING PERFORMANCE DATA

The CPOM double pilot operated check valve blocks leakage from the actuator ports to tank when the directional valve is in the center position.

**NOTE:** For maximum response and shut off, a directional valve with both cylinder ports drained to tank in the center position is recommended for use with MANAPAK double pilot check valves.

#### MAXIMUM OPERATING PRESSURE

3000 PSI (205 Bar) — CPOM3 & CPOM6  
4500 PSI (307 Bar) — CPOM2

#### NOMINAL FLOW

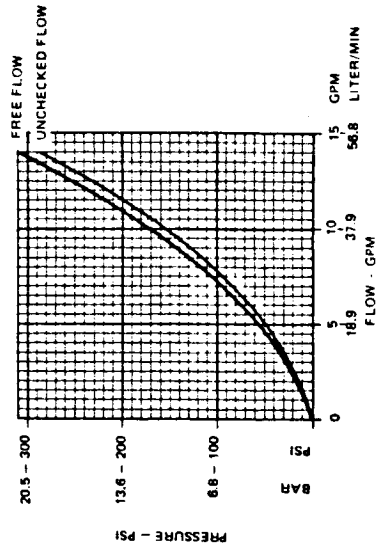
CPOM2 7 GPM (26 LITER/MIN.)  
CPOM3 12 GPM (45 LITER/MIN.)  
CPOM6 50 GPM (189 LITER/MIN.)

#### CRACKING PRESSURE

5 PSI (.34 Bar)

QUICK REFERENCE DATA CHART				
Valve Model	Flow (Maximum)	P.P. Maximum Free Flow	Pilot Piston Check Valve Area	Use with Directional Valve Series
CPOM2	14 GPM (53 L/M)	305 PSI (21 Bar)	3.1	DIV
CPOM6	80 GPM (227 L/M)	350 PSI (24 Bar)	3.1	D8

CPOM2



VISCOSITY CORRECTION FACTOR				
Viscosity (SSU)	75	150	200	250
Percentage of $\Delta P$ (Approx.)	93	111	119	126
Drop will change as per chart.	137	141		

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## Technical Information

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3000 PSI (205 Bar) — CPOM3 & CPOM6  
4500 PSI (307 Bar) — CPOM2

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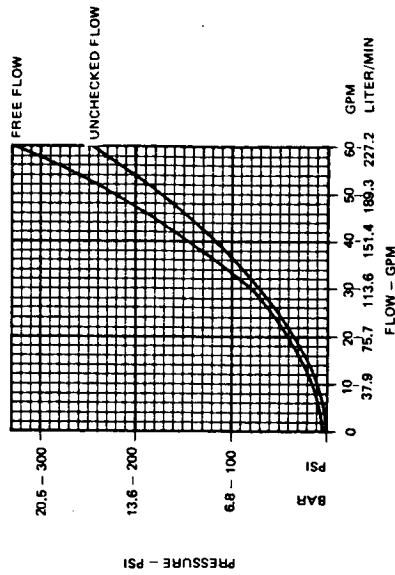
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CPOM2	14 GPM (53 L/M)	305 PSI (21 Bar)	3.1	DIV
CPOM6	80 GPM (227 L/M)	350 PSI (24 Bar)	3.1	D8

CPOM6



VISCOSITY CORRECTION FACTOR				
Viscosity (SSU)	75	150	200	250
Percentage of $\Delta P$ (Approx.)	93	111	119	126
Drop will change as per chart.	137	141		

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## Series CPOM

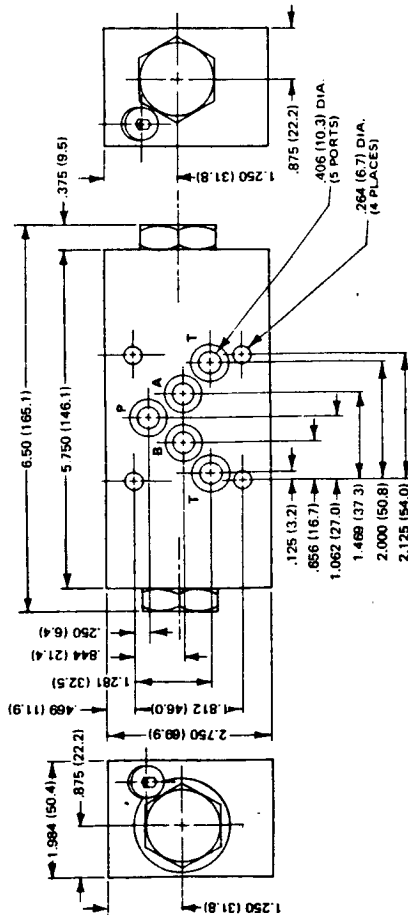
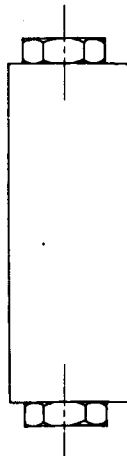
Manapak, Pilot Operated  
Valves, Check

## Technical Information

### DIMENSIONS

\*MILLIMETER EQUIVALENTS FOR INCH DIMENSIONS ARE SHOWN IN (\*\*)

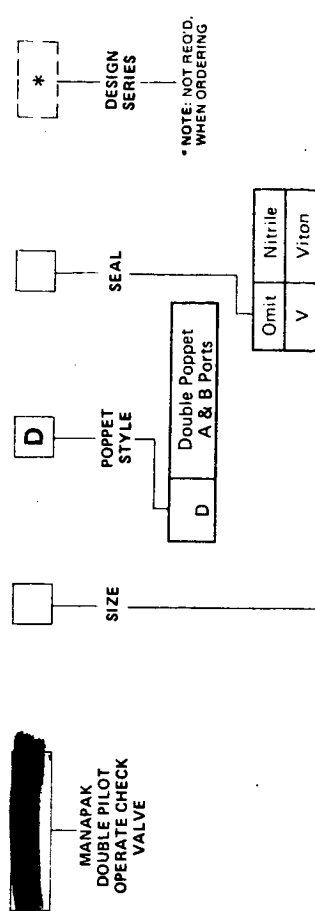
MANAPAK-PILOT OPERATED-CHECK VALVE  
USED WITH D3 DIRECTIONAL VALVE



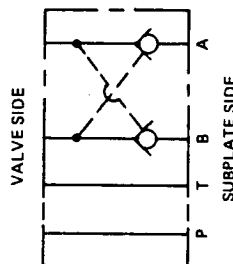
## Series CPOM

Manapak, Pilot Operated  
Valves, Check

## Ordering Information



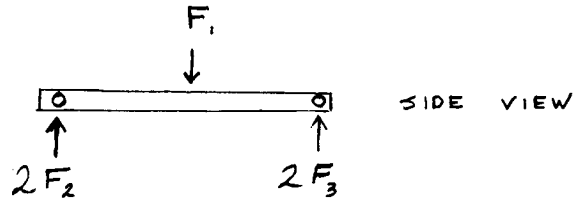
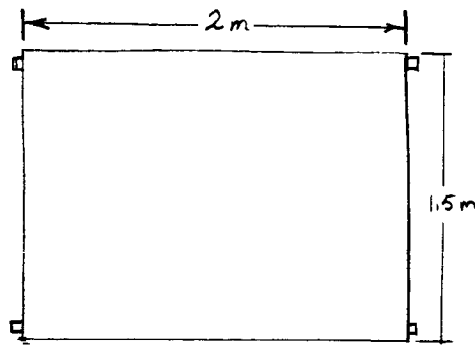
2	NFPA DO1 Subplate Mounting
6	NFPA DO6 Subplate Mounting



UNIT WEIGHT:	VALVE	WEIGHT
CPOM2D		1.7 LBS. (.7 Kg)
CPOM3D		9.6 LBS. (4.4 Kg)
CPOM6D		21.0 LBS. (9.5 Kg)

# DOOR HITCH

1



Assuming Lunar Soil acts as a fluid which is the worst case.

$$F_1 = \rho g h_{CG} A$$

$$A = (1.5)(2.0) = 3.0 \text{ m}^2$$

$$h_{CG} = 0.75 \text{ m}$$

$$g = \frac{1}{6}(9.81) = 1.64 \text{ m/s}^2$$

$$\rho = 1.7 \text{ g/cm}^3$$

$$\therefore F_1 = (1.7)(1.64)(0.75)(3.0) = 6.25 \frac{\text{m}^4 \cdot \text{g}}{\text{cm}^3 \cdot \text{s}^2} \left( [1000]^3 \frac{\text{cm}^3}{\text{m}^3} \right) \left( \frac{1}{1000} \frac{\text{kg}}{\text{g}} \right)$$

$$F_1 = 6,250 \text{ N}$$

With an acceleration of  $5 \text{ m/s}^2$

$$P_A = \rho G h$$

$$G = [g^2 + a^2]^{1/2} = [(1.64)^2 + (5)^2]^{1/2} = 5.26 \text{ m/s}^2$$

$$P_A = (1.7)(5.26)(1.5) = 13.4 \text{ N/m}^2$$

$$\text{New } F_1 = (P_A + \rho g h_{CG}) A$$

$$= (13.4 + 2,080) 3.0$$

$$F_1 = 6280 \text{ N}$$

NEXT CONSIDERATION:

How large a rock will exert a force of  $F_1$  if it were to roll from the opposite high corner to the corner of the door hitch?

This can be simplified if one assumes that this fall is the same as free fall from the height of the loader bucket. This will be approximately 2.4 m

$$F_1 = 6280 \frac{\text{kg} \cdot \text{m}}{\text{s}^2}$$

$$\text{Mass of rock} = \frac{F_1}{g} = \frac{6280}{1.64} = 3830 \text{ kg}$$

Considering that the trailer will carry 15,800 kg of soil.

A rock of the size of 3830 kg is 25% of the entire load.

A rock this big would be broken-up before it was placed in the trailer.

FORCES UPON PINS:

By STATICS

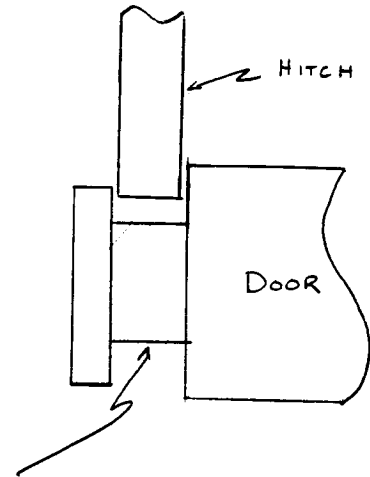
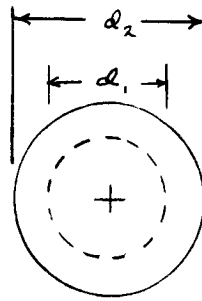
$$\Sigma F = 0 = 2F_2 + 2F_3 = 6280$$

$$\Sigma M_3 = 0 = 0.75(6280) = 2(1.5)F_2$$

$$\therefore F_2 = 1,570 \text{ N}$$

$$F_3 = 1,570 \text{ N}$$

THEREFORE a pin and hitch that can withstand a load of 1,570 N is needed.

PIN DESIGN

PIN: UNDER  
ANALYSIS

SHEAR STRESS

$$\tau = \frac{F}{A}$$

$$F = n F_2$$

$$A = \frac{\pi d_1^2}{4}$$

$$\tau = 0.577 S_y$$

Choosing An Aluminum Alloy

$$A92219 - T89 \Rightarrow S_y = 50(10)^3 \text{ psi} \\ = 345 \text{ MPa}$$

so with  $n = 1.5$  (Low since soil is not a fluid)

$$d = \left[ \frac{4 n F_2}{\pi 0.577 S_y} \right]^{1/2} = \left[ \frac{4(1.5)(1570)}{\pi 0.577 (345)(10)^6} \right]^{1/2}$$

$$d = 3.88(10)^{-3} \text{ m}$$

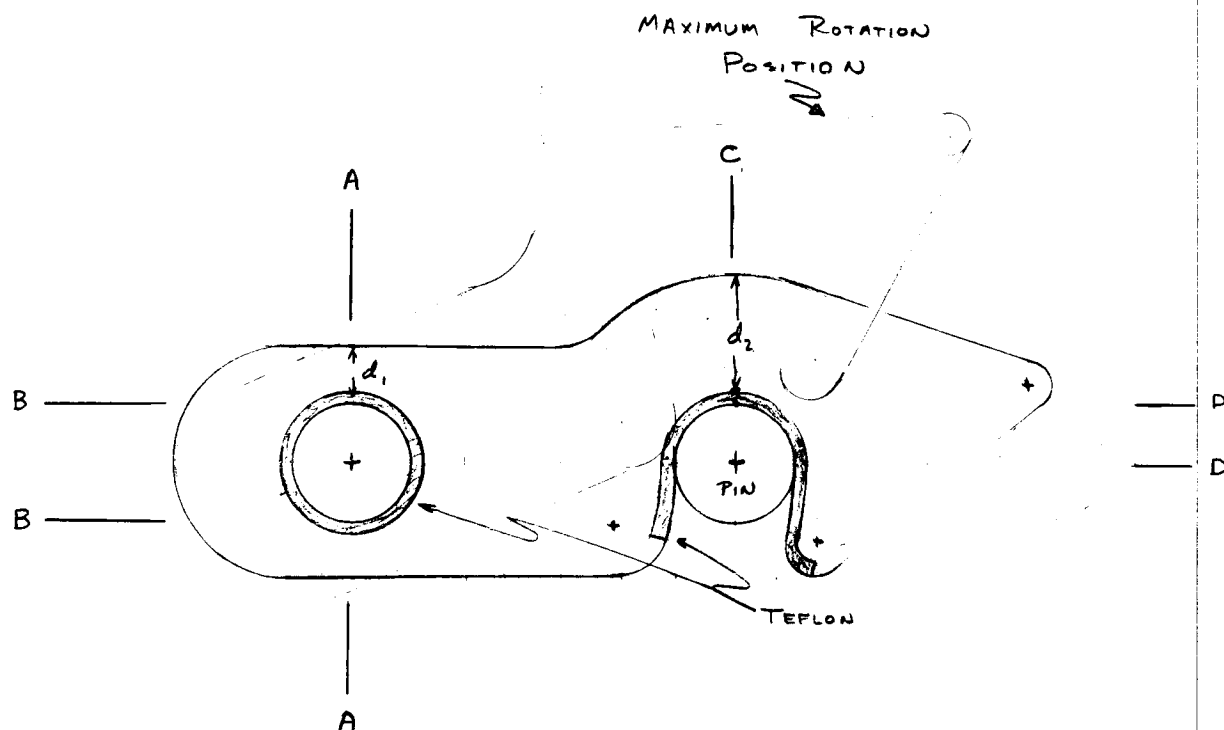
$$d = 0.388 \text{ cm}$$

If  $d = 1.3 \text{ cm}$

$$n = 16.8$$

Therefore, a pin of 1.3 cm in diameter has a safety factor of at least 6.7

### HITCH DESIGN :



### SHEAR STRESS ACROSS PLANE A

$$A = \frac{F}{\tau}$$

$$F = 11570 \text{ N}$$

$$\tau = 0.577 S_y$$

$$A = 2d_1 t \quad t (\text{thickness}) = 0.8 \text{ cm}$$

$$\therefore d_1 = \frac{11570}{0.577(345)10^6(2)(0.008)}$$

$$d_1 = 0.35 \text{ cm}$$

### BEARING STRESS AT B PLANE

$$\sigma_y = \frac{n F}{A}$$

$$\sigma_y = 138 \times 10^6 \text{ Pa}$$

$$nF = 6.7(1570) \text{ N}$$

$$A = 1.3 t$$

$$\therefore t = \frac{7.0(1570)}{345(10)^6 (1.3)}$$

$$t = 0.002 \text{ cm} \quad \therefore t = 0.8 \text{ cm has } n = 2285$$

### BEARING STRESS AT D PLANE

$$n = \frac{A \sigma_y}{F}$$

$$A = \frac{1.3}{2} t$$

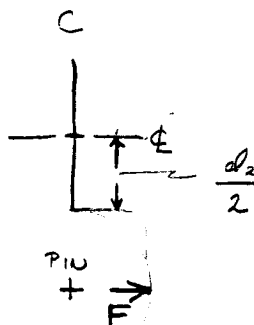
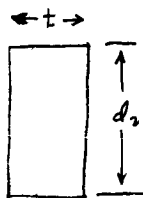
$$F = 1570 \text{ N}$$

$$\therefore n = \frac{(1.3)(0.008)(345(10)^6)}{(1570) 2}$$

$$\sigma_y = 345 \times 10^6 \text{ Pa}$$

$$n = 1143$$

### BENDING AT C PLANE



### Bending Moment

$$M = \frac{1}{2} F (\phi \phi 13 + d_2)$$

$$C = \frac{d_2}{2}$$

$$I = \frac{bh^3}{12} = \frac{\pm d_2^3}{12}$$

$$\sigma = \tau_y = 345 \times 10^6 \text{ Pa}$$

Minimum  $d_2$  ✕

$$\frac{\sigma}{n} = \frac{M_c}{I} = \frac{c}{I} M = \frac{\frac{d_2}{2}}{\frac{\pm d_2^3}{12}} \left[ \frac{1}{2} F (\phi \cdot \phi 13 + d_2) \right]$$

$$\frac{\sigma}{n} = \frac{3F}{\pm d_2^2} \left( \phi \phi 13 + \frac{d_2}{2} \right)$$

$$\frac{\sigma}{n} = \frac{\phi \cdot \phi 39 F}{\pm d_2^2} + \frac{1.5 F}{\pm}$$

$$\frac{1}{d_2^2} = \frac{t}{\phi \cdot \phi 39 F} \left[ \frac{\sigma}{n} - \frac{1.5 F}{t} \right]$$

$$\frac{1}{d_2^2} = \frac{\phi \cdot \phi 08}{\phi \cdot \phi 39 (157 \phi)} \left[ \frac{345 \times 10^6}{7.0} - \frac{1.5 (157 \phi)}{\phi \cdot \phi 08} \right]$$

$$d_2^2 = 1.56 \text{ cm}^2$$

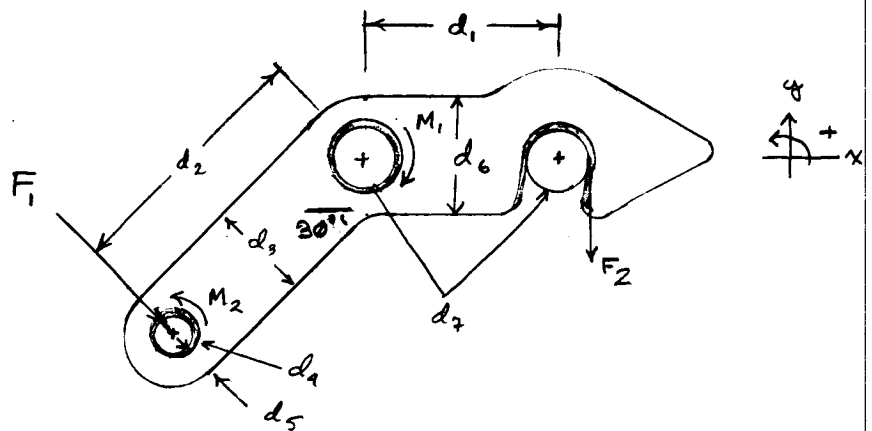
$$\therefore d_2 = 1.25 \text{ cm}$$

## SOLINOID PLACEMENT

The static coefficient of friction ( $\mu_s$ ) between aluminum and teflon is approximately equal to 0.04. Therefore, the solinoid needs to overcome three points of friction:

- 1) Force at Pin & Hook
- 2) MOMENT FORCE at rotation pin
- 3) MOMENT FORCE at push rotation pin

$$\begin{aligned}
 d_1 &= 5.5 \text{ cm} \\
 d_2 &= 10.0 \text{ cm} \\
 d_3 &= 0.80 \text{ cm} \\
 d_4 &= 0.32 \text{ cm} \\
 d_5 &= 0.44 \text{ cm} \\
 d_6 &= 3.36 \text{ cm} \\
 d_7 &= 1.3 \text{ cm} \\
 t &= 0.2 \text{ cm} \\
 \text{TEFLON}
 \end{aligned}$$



The choice of  $d_1$  is fairly arbitrary. The only criteria is buckling, but this is not a problem is  $d_1$  is not much larger than  $d_6 = 3.36 \text{ cm}$  with teflon thickness of  $0.2 \text{ cm}$ . I choose  $5.5 \text{ cm}$  for  $d_1$ .

### STATIC FORCES

$$F_2 = d_1 \mu_F F_{\text{DOOR}} = (5.5)(0.04)(1570)$$

$$F_2 = 3.45 \text{ N} \cdot \text{m}$$

$$M_1 = \mu_F F_{\text{DOOR}} \left( \frac{\pi d_7}{2} + t \right) = 0.04 (1570) \left( \frac{\pi (0.013)}{2} + 0.008 \right)$$

$$M_1 = 1.78 \text{ N} \cdot \text{m}$$

STATIC FORCES (CONT)

$$M_2 = \mu_F F_1 \left( \frac{\pi d_4}{2} + t \right)$$

$$F_1 = \frac{1}{d_2} [F_2 + M_1 + M_2] = \frac{\pi d_4 \mu_F}{d_2} F_1 + \frac{1}{d_2} [F_2 + M_1] + \mu_F F_1 t$$

$$\therefore F_1 = \frac{1}{d_2} \left( 1 - \frac{\pi d_4 \mu_F}{d_2} - \mu_F t \right)^{-1} [F_2 + M_1]$$

Choosing arbitrary,  $d_2 = 7.0 \text{ cm}$   
 $d_4 = 0.75 \text{ cm}$  } checked later

$$F_1 = \frac{1}{0.07} \left( 1 - \frac{\pi(0.0075)(0.07)}{0.07} - (0.04)(0.008) \right)^{-1} [3.45 + 1.78]$$

$$F_1 = 75.8 \text{ N}$$

Choosing  $d_2 = 10.0 \text{ cm}$

$$F_1 = 52.8 \text{ N}$$

Therefore, a force of

$$F_1 = 60.0 \text{ N will suffice}$$

3.5 mm

BENDING ON  $d_3$ 

$$\frac{\sigma}{n} = \frac{C}{I} M = \frac{\frac{d_3}{2}}{\frac{\pi d_3^4}{12}} F_1 d_2$$

$$\frac{\sigma}{n} = \frac{6}{\pi d_3^3} F_1 d_2$$

$$d_3^2 = \frac{6 \pi F_1 d_2}{\pi \sigma} = \frac{6(7)(60)(0.07)}{(0.008) 345(10^6)}$$

$$d_3 = 0.80 \text{ cm}$$

PIN DIAMETER  $d_4$

$$\tau = \frac{F}{A}$$

$$F = \pi F_z$$

$$A = \frac{\pi d_4^2}{4}$$

$$\tau = 0.577 S_y$$

$$d_4 = \left[ \frac{4 \pi F}{\pi 0.577 S_y} \right]^{1/2} = \left[ \frac{4 (7.0) 6.0}{\pi 0.577 (345 \times 10^6)} \right]^{1/2}$$

$$d_4 = 0.16 \text{ cm}$$

BEARING STRESS FOR  $d_5$

$$d_5 = \frac{\pi F}{t S_y} = \frac{7.0 (6.0)}{0.008 (345 \times 10^6)}$$

$$d_5 = 0.02 \text{ cm}$$

$$\text{Since } d_3 = 0.80 \text{ cm}$$

$$\begin{aligned} \text{make } d_5 &= 0.44 \text{ cm} \\ d_4 &= 0.32 \text{ cm} \end{aligned} \quad \text{w/ Teslon } t = 0.2 \text{ cm}$$

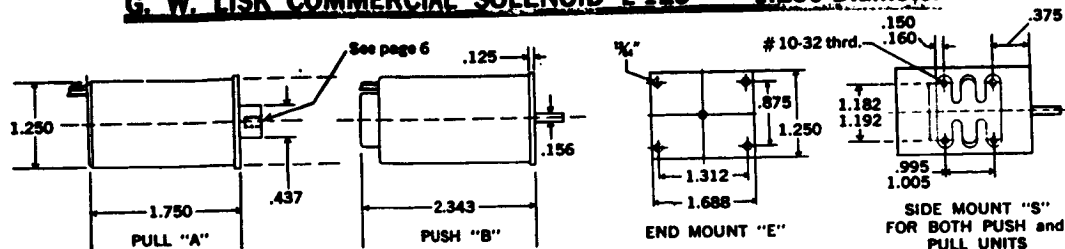
SOLENOID USED

Solenoid used is made by GW Lisk Co.

Part No.

L-125 BE 60 K3 L1 24 I

DESCRIPTION OF PART IN APPENDIX

**G. W. LISK COMMERCIAL SOLENOID L-125 1.250 Diameter**

Note: Mounting foot thickness is .125

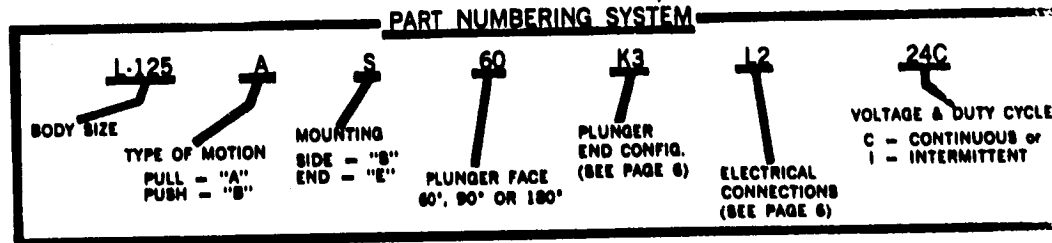
**CONTINUOUS DUTY**

CONTINUED FROM

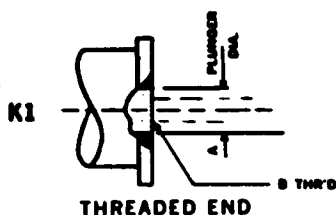
POLE FACE	NOM. COIL POWER	% COIL VOLTS	AVERAGE SOLENOID FORCE IN POUNDS—NOMINAL STROKE IN INCHES																		
			.000"	.010"	.020"	.030"	.040"	.052"	.063"	.125"	.156"	.187"	.250"	.312"	.375"	.437"	.500"	.625"	.750"	.875"	1.000"
1 (60°)	12.6	85%	9.3	6.7	6.1	5.1	4.6	3.4	2.5	1.8	1.3	1	.6	.4	.3	.2	.1				
	watts	100%	11	9	8.2	6.8	5.8	4.5	3.1	2.4	1.8	1.4	.8	.6	.4	.3	.2				
2 (90°)	12.6	85%	14.2	11.9	9.2	6.9	5.2	3.4	2.1	1	.7	.6	.4	.2	.1	.1					
	watts	100%	15.5	12.6	10.2	8.1	6.3	4.4	2.7	1.4	1.2	.9	.6	.4	.3	.2	.1				
3 (180°)	12.6	85%	29.8	16.1	10.6	7.2	5.1	2.3	1.4	.8	.5	.3	.2	.1	.1						
	watts	100%	31.5	17.6	12.4	8.9	6.7	2.9	1.9	1	.8	.5	.3	.2	.2						

**INTERMITTENT DUTY**

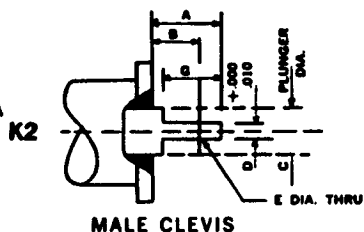
INTERMITTENT DUTY																		
POLE FACE STYLE	NOM. COIL POWER	% COIL VOLTS	AVERAGE SOLENOID FORCE IN POUNDS—NOMINAL STROKE IN INCHES															DUTY CYCLE
			.000"	.020"	.032"	.063"	.125"	.156"	.187"	.250"	.375"	.500"	.625"	.750"	.875"	1.000"		
1 (60°)	240	85%	16.2	14	11.7	10.9	10.2	9.8	9.2	8.2	6.8	5.2						6% MAX "ON" TIME = 1 min
	watts	100%	17	14.8	12.4	11.6	10.9	10.4	9.8	8.9	7.9	6.6						
2 (90°)	240	85%	21.9	16.9	14.7	13.1	11.8	10.4	9.2	7.2	4.3	2.5						
	watts	100%	23	18	15.6	14	12.6	11.4	10.2	8.2	5.5	3.5						
3 (180°)	240	85%	36.6	24.9	18	13.2	10.3	8.6	7.4	5.6	3.7	2.1						
	watts	100%	38.5	26.6	19.4	14.5	11.6	9.9	8.6	7	5.1	3.2						

**PART NUMBERING SYSTEM****NOTES:**

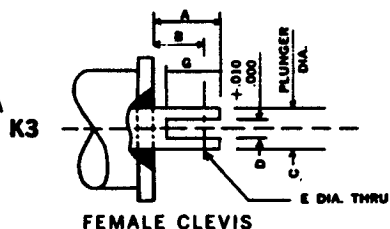
1. FOR INTERMITTENT DUTY, COILS ARE AVAILABLE WITH LOWER POWER CONSUMPTION, LONGER "ON" TIME WITH LESS FORCE. FOR INFORMATION CONTACT G. W. LISK SALES DEPT.
2. FOR PLUNGER END CONFIGURATION SEE PAGE 6.
3. FOR ELECTRICAL CONNECTION SEE PAGE 6.
4. MAX. STABILIZED COIL TEMP. AT AMBIENT OF 76°F IS 220°F.

**PLUNGER END CONFIGURATION—SHOWN ENERGIZED**PULL  
TYPE A

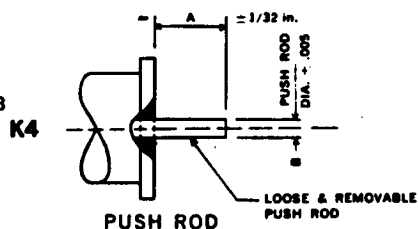
SIZE	A DIA.	B THRD	
L-125	.437	± 6-32 .250 MIN.	
L-162	.625	± 8-32 .312 MIN.	
L-200	.750	1 4-28 .375 MIN.	
L-250	.937	1 4-28 .375 MIN.	
L-300	1.250	5/16-18 .625 MIN.	

PULL  
TYPE A

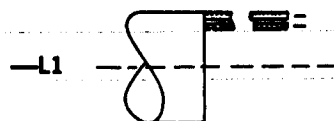
SIZE	A	B	C DIA.	D	E DIA.	F	G
L-125	.625	.437	.437	.125	124 129		.500
L-162	.625	.437	.625	.156	124 129		.500
L-200	.750	.500	.750	.250	155 161		.625
L-250	.875	.562	.937	.250	249 255		.750
L-300	.937	.625	1.250	.312	249 255		.812

PULL  
TYPE A

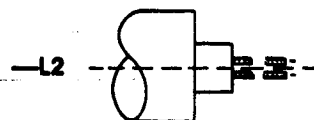
SIZE	A	B	C DIA.	D	E DIA.	F	G
L-125	.625	.437	.437	.125	124 129		.500
L-162	.625	.437	.625	.156	124 129		.500
L-200	.750	.500	.750	.250	155 161		.625
L-250	.875	.562	.937	.250	249 255		.750
L-300	.937	.625	1.250	.312	249 255		.812

PUSH  
TYPE B

SIZE	A	B	C	D	E	F	G
L-125	To Suit	.156					
L-162	To Suit	.203					
L-200	To Suit	.250					
L-250	To Suit	.250					
L-300	To Suit	.250					

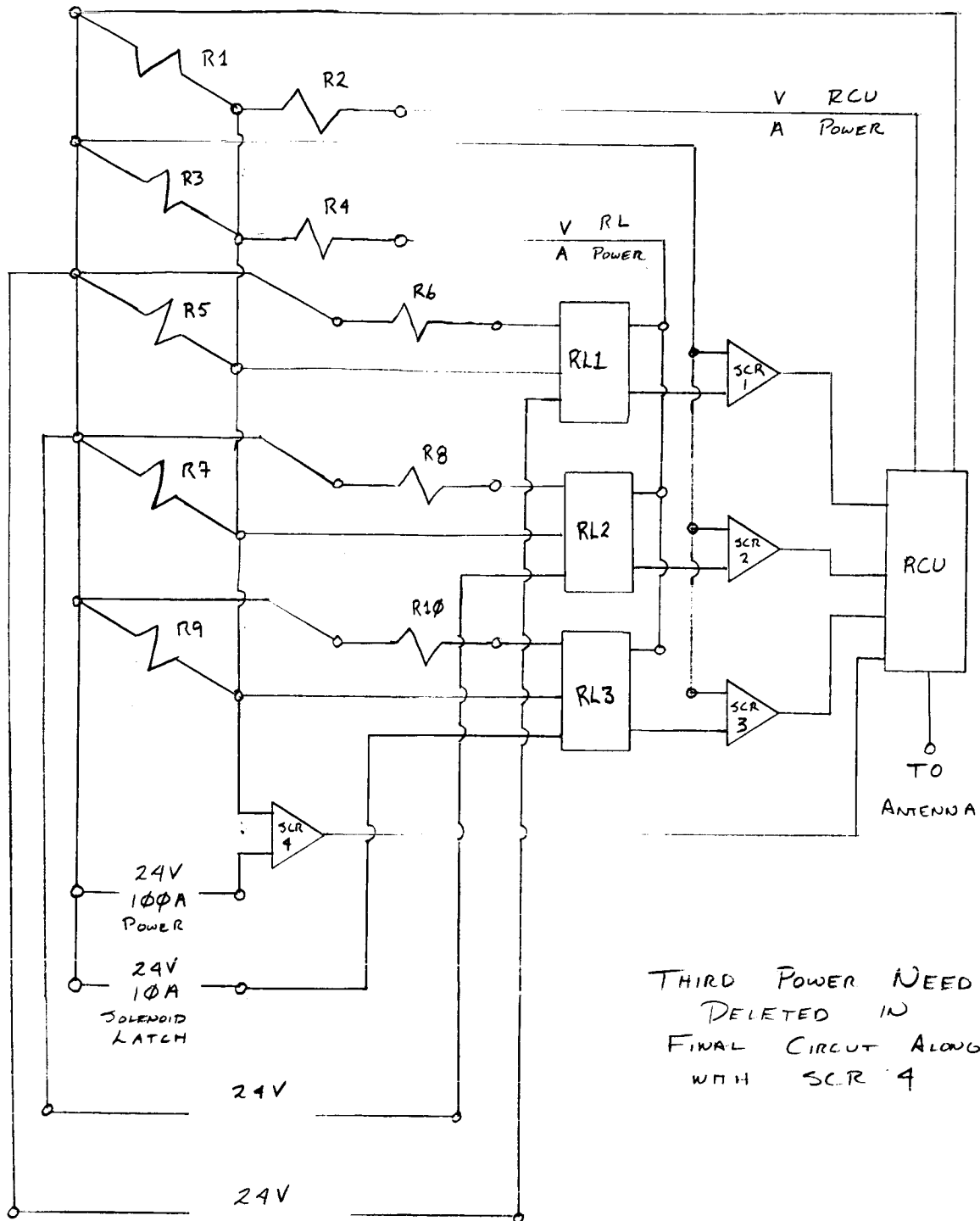
**ELECTRICAL CONNECTION**

12" LEADWIRES

12" LEADWIRES THRU  
1/2" PIPE LINE CONDUIT

# REMOTE CONTROL

1



THIRD POWER NEED  
DELETED IN  
FINAL CIRCUIT ALONG  
WITH SCR 4

RESISTOR DETERMINATION

$$\text{CHOOSE } R_6 = 0.75 \Omega \Rightarrow 770 \text{ W} \quad [P = I^2 R]$$

$$R_4 = 0.75 \Omega \Rightarrow 770 \text{ W}$$

$R_4$  &  $R_6$  would dissipate 32 A each

THEREFORE:

$$\text{PNT D has } 100 - 32 - 20 = 48 \text{ A leave}$$

$$\text{PNT C has } 48 - 32 - 1.5 = 14.5 \text{ A leave}$$

$$\text{CHOOSE } R_3 = 2.25 \Omega$$

$$R_3 \text{ would dissipate } 10.7 \text{ A} \Rightarrow 256 \text{ W}$$

THEREFORE:

$$\text{PNT B has } 14.5 - 10.7 - 2 = 1.8 \text{ A leave}$$

AS A RESULT:

$$R_1 = \frac{V}{I} = \frac{24}{1.8 - 0.01} = 13.4 \Omega \Rightarrow 44 \text{ W}$$

FOR  $R_2$

$$R_2 = \frac{V}{I} = \frac{1.5 \text{ V}}{10 \text{ mA}} = 150 \Omega \Rightarrow 0.5 \text{ W}$$

FOR  $R_5$

$$R_5 = \frac{V}{I} = \frac{24}{1.5} = 16 \Omega \Rightarrow 36 \text{ W}$$

FOR  $R_7$

$$R_7 = \frac{V}{I} = \frac{24}{20} = 1.2 \Omega \Rightarrow 480 \text{ W}$$

## Radio Control Modules

Ideal for wireless remote control of toys, models, novelty items and more. These prewired PC modules require just a few low-cost components to complete your system. Provides four functions: Forward, forward with right turn, forward with left turn and reverse. Receiver controls two DC motors directly. Transmitter,  $3\frac{11}{16} \times 1 \times \frac{7}{16}$ " Receiver,  $1\frac{5}{8} \times 3 \times \frac{3}{4}$ ". Operate on 27.145 Mhz. Require 9VDC and 1.5VDC. With data.

277-1012

Set, 16.95

NEW  
FOR '84

16.95  
Set

Receiver



Transmitter



## PARTS LIST

The following is a comprehensive list of all the parts used in the construction of the lunar dump trailer.

### BED ASSEMBLY

- 1 cast aluminum trailer bed
- 23 meters aluminum T-beam (see specifications)
- 2 machined aluminum brackets (see drawing)
- 2 hinge bolts (door) with nuts
- 1 cast aluminum bed door
- 2 pivot pins (68371)
- 1 clevis bracket (69208)
- 8 bolts (912024-T4, 5/8in dia, 2in length)
- 8 nuts (5/8in Hthreads, CNP-3)

### LATCH ASSEMBLY

- 2 machined aluminum latches
- 2 sheet aluminum latch covers
- 2 latch solenoids (L-125 BE60K3L124I)
- 8 teflon coated washers
- 4 cover hinges
- 8 nuts (see specifications)
- 2 safety labels (see drawing)

### REMOTE CONTROL ASSEMBLY

- 1 remote control assembly
- 1 remote control antenna
- 1 radio receiver (277-1012)
- 2 0.75 ohm, 770 watt resistors
- 1 2.25 ohm, 256 watt resistor
- 1 13.4 ohm, 44 watt resistor
- 1 150 ohm, 0.5 watt resistor
- 1 16 ohm, 36 watt resistor
- 1 1.2 ohm, 480 watt resistor
- 2 relays
- 2 SCR's

- 1 box
- 3 meters coaxial antenna cable
- 15 meters wire
- assorted attachment hardware

#### WHEEL ASSEMBLY

- 2 forged AL-95056-0 trailer wheel
- 2 double roller tapered bearings
- 2 lug nuts
- 2 cotter pins

#### REAR SUSPENSION ASSEMBLY

- 3 meter stock steel, machined
- 2 hub assemblies
- 2 axle assemblies
- 6 attachment pins
- 6 retainer pins
- 2 springs
- 2 dampers
- 4 spring retainer brackets
- 8 pin brackets
- 8 bolts and nuts
- 2 machined steel plates

#### FRAME ASSEMBLY

- 15 meter aluminum I-beam (see specifications)
- 74 bolts (1 1/4 - 8 thread 5 in) with matching bolts
- 8 L-beam bolts
- 1 hitch

#### HYDRAULIC SYSTEM

- 1 hydraulic cylinder (4.00CBB2H24CX28)
- 1 flow control valve (FS1200)
- 1 solenoid operated control valve (D3W4CJ)
- 1 subplate (SPD34)
- 1 pilot-operated check valve (CPOM3D)
- 1 mounting plate (69198)
- 1 eye bracket (69200)
- 1 pivot pin (69215)
- 4 bolts (21/32 8 thread 4.5 in) with matching nuts
- 4 bolts (1 1/16 8 thread 5 in) with matching nuts
- 10 meters hydraulic hose (Parker No-skive, .870in I.D 1in O.D)

2 double shut-off quick connect couples (60 series)

ME 4182: WEEKLY REPORT  
4 APRIL - 11 APRIL

This week was spent in the library looking for resources relative to the moon's environment. Some books were checked out and others noted. Not much else was done since we were not exactly sure what we were required to do with the resources. We did not know if the sources were to be put on reserve or if information was to be extracted and then put on reserve.

**ME 4182: WEEKLY REPORT**  
**11 APRIL - 18 APRIL**

This week was also spent in the library looking for resources relative to the moon's environment. Of the books found, the information relative to the moons environment was noted. A synopsis of each source was compiled and recorded for the ME 4182 file.

We also came up with a number of different methods of designing the lunar dump truck. A general design of a soft material carrier supported by metal or plastic tubes was decided upon as our first approach. The method considered of removing the material from the truck was that of splitting the load space and allowing gravity to remove the material. Gearing motorized from the towing vehicle is our present thought on the opening and closing of the dump truck. This is all the brainstorming that we accomplished with the information that we presently have at our disposal.

**ME 4182: WEEKLY REPORT**  
**18 APRIL - 25 APRIL**

This week was the final week spent in the library looking for resources relative to the moon's environment. At the completion of the week, all of the data compiled over the weeks was placed in the ME 4182 information sources book. We placed it in the wrong place within the book but it was in the notebook.

Our group decided during the week to design a square dump truck with four wheels. It will lift by hydraulics in a manner similar to that of an earth dump truck. The door will be controlled by an electronic latch.

ME 4182: WEEKLY REPORT  
25 APRIL - 2 MAY

At this time we divided the project into several different areas. Each respective area was assigned to a person in the group. The areas that were subdivided with their respective designer are: bucket and door, Soupiset; cover and report intro, Farkas; electronics and remote control, Hicks; frame, Johnson; wheels and suspension, Bade; towing, Skinner. We decided that the next few weeks would be spent designing our respective areas with meetings to discuss each others designs. Problems that arise and interaction of designs would also be discussed.

ME 4182: WEEKLY REPORT  
2 MAY - 9 MAY

Each respective member worked on his area of the lunar dump truck. The wheel design is beginning to look like the most work. Therefore Skinner and Bade have combined effort. They are trying to figure out have to get a smooth ride along with proper steering. The other subdivision of the project are not running into any real difficulties at this time.

ME 4182: WEEKLY REPORT  
9 MAY - 16 MAY

By this week, most of the major design decisions had been made. A two wheel design is being used for ease of steering and complexity. The latch is being controlled by a solenoid. The hydraulic system is very similar to that of an earth dump truck. The material for the body and frame was chosen. The design of the bucket was in the final stages. The frame had not been looked at due to it interacting with the bucket and suspension which were not completed.

**ME 4182: WEEKLY REPORT**  
**16 MAY - 23 MAY**

At this point in time, the bucket is virtually done along with the dust cover. The wheel suspension system is in its final stages along with the latch component. Much difficulty is arising in the location of a remote control unit however. The hydraulic system has also been decided upon and needs to be put together. At this point the frame is ready to be designed. The entire design process is in the near final stages and a report write up is being considered.

ME 4182: WEEKLY REPORT  
23 MAY - 30 MAY

Surprise! A good description for this week. We found out tuesday that the rough draft is due at the end of this week. We did not think the report was due till monday. Therefore its time to cram one whole week into half of one. This results in the completion of the project except for the final draft to be compiled over the weekend. Under the condition that the frame is started and designed.

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## ACKNOWLEDGEMENTS

Bade	Wheel Design Drawings 1, 2, 7, 9, Drawings 10, 11, 12 Operating Instructions Maintenance
Farkas	Abstract Constraints Trailer Hitch
Hicks	Drawings 14, 15, 16, Drawings 17, 18 Door Hitch Remote Control Typing
Johnson	Frame Design Remote Control
Skinner	Suspension Drawing 8 Conclusion
Soupiset	Bed Design Typing Drawings 3, 4, 5, 6
Stubbs	Hydraulic Design Drawing 13 Performance Objectives